THE INTERNAL COMBUSTION ENGINE

WIMPERIS

THE INTERNAL COMBUSTION ENGINE

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The Internal Combustion Engine

A Text-Book for the Use of Students and Engineers

By

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EXTRACT FROM PREFACE TO FIRST EDITION

The present book deals with subjects in the borderland between several allied sciences (notably physics and chemistry) and the exclusively practical sides of their application. It is hoped that the student will be helped to understand something of the applications of those heat engines which work on the internal combustion principle, and the engineer to a fuller realization of the scientific principles concerned in the design and working of gas, oil and petrol engines. In order to economize space, and since it has been amply dealt with by many other writers, little is said of the historical side of the subject. The introduction, into the theoretical treatment of the subject, of the principle of the now recognized variability of specific heats with temperature has involved the breaking of much new ground, and it is impossible to expect complete success in avoiding mistakes and slips in the calculations. I shall therefore be very glad to have brought to my notice any corrections that may be necessary.

In writing this book so many original papers and treatises have had to be consulted that it is not easy to make the requisite and proper acknowledgments. First, however, it is a pleasure to acknowledge my great indebtedness to Professor Perry, to whom, as a student many years ago, and on numerous occasions since, my thanks are due for guidance, counsel and help generously given. I have also to thank Mr. Dugald Clerk and Professor Hopkinson for copies of their papers. I am indebted also to Mr. J. T. H. Burrell, Assoc.M.Inst.C.E., for checking the mathematical calculations and for working out the examples. For the illustrative matter I have to thank the Institutions, Firms and

individuals mentioned in the following list, but chiefly my friend Mr. F. Strickland and Messrs. Chas. Griffin & Co. for permission to reproduce certain illustrations from their "Petrol Motors and Motor Cars." Finally I tender my thanks to the Editors of *The Engineer* and *Engineering* for permission to reproduce certain parts of articles contributed to their columns.

H. E. W.

CHELSEA
13*th* *August* 1908

PREFACE TO SECOND EDITION

Since the First Edition of this book was printed there have been many important developments in the internal combustion engine. There has been a considerable extension of its use at sea, a very largely increased employment on land, and a most notable development for service in the air. Moreover, the recent work of the B.A. Gaseous Explosions Committee has provided a real basis for a modern theory of the engine. These changes in theory and practice have necessitated corresponding changes in the book; the new matter to be added has led to some compression in the old, so that the length might be kept within bounds: much has been rewritten. In this work I have received the valuable assistance of Mr. H. E. Piggott, M.A., formerly scholar of Clare College, Cambridge, and now Head of the Mathematical Department of the R.N. College, Dartmouth, and of Mr. W. E. Hogg, A.R.C.S., Assoc.M.Inst. C.E.; to both of them my thanks are due.

At the end of each chapter will be found some problems for solution, drawn chiefly from the examination papers set at Cambridge, at the Imperial College of Science and Technology, and by the Board of Education and other Government Departments. These have been arranged by Mr. Piggott, and his solutions of them are given at the end of the book.

H. E. W.

Hampstead
28th February 1915

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LIST OF CHIEF SYMBOLS USED

P = Pressure
V = Volume
T = Temperature absolute (Centigrade)
θ = Temperature (Centigrade) as read by thermometer
E = Internal energy
φ = Entropy
v = Velocity
C_p = Specific heat at constant pressure
C_v = Specific heat at constant volume
J = Mechanical equivalent of heat or "Joule's Equivalent"
t = Time in seconds
η = Efficiency
$\gamma = \frac{C_p}{C_v}$
$R = C_p - C_v$
ω = Angular velocity
I = Moment of Inertia
I.H.P. = Indicated Horse-Power
B.H.P. = Brake Horse-Power
kW = Kilowatts
r = Compression-ratio

USEFUL CONSTANTS

Length, Area and Volume

1 centimetre = 0·3937 inch.
1 inch = 2·540 cm.
1 sq. cm. = 0·1550 sq. in.
1 sq. in. = 6·452 sq. cm.
1 cu. metre = 35·31 cu. ft.
1 imperial gallon = 4·546 litres = 10 lb. of water.
1 U.S.A. gallon = 3·785 litres.

Weight and Pressure

1 kg. = 2·205 lb.
g = 32·2 ft. per sec. per sec. = 981 cm. per sec. per sec.
1 atmosphere = 14·7 lb. per sq. in. = 760 mm. of mercury
= 2,116 lb. per sq. ft. = 34 ft. of water.
1 lb. per sq. in. = 0·07031 kg. per sq. cm.
1 litre of water weighs 1 kg. = 1000 grams.
1 metric ton = 2205 lb.

Energy

1 ft. lb. = 0·1383 kg. metre = 1·356 × 10^7 ergs.
1 Joule = 10^7 ergs = 0·7373 ft. lb.
1 H.P.-hour = 1,980,000 ft. lb.
1 C.H.U. = 1,400 ft. lb.
1 B.Th.U. = 778 ft. lb.
1 calorie = 3,087 ft. lb. = 2·205 C.H.U.

Power

1 watt = 1 volt. × 1 ampere = 10^7 ergs. per sec. = 1 Joule per sec.
1 K.W. = 1·34 H.P. = 0·239 calories per sec.
1 H.P. = 0·746 K.W. = 76·04 kg. m. per sec.
1 metric H.P. = 0·986 English H.P. = 75 kg. m. per sec.

OTHER CONSTANTS

1 cu. ft. of water = 62·3 lb.
1 cu. ft. of air (N.T.P.) = 0·0807 lb.
1 radian = 57·3 deg.
$\log_e x = 2{\cdot}3026 \times \log_{10} x$.
$e = 2{\cdot}7183$.
Absolute zero = −273° C. = − 459° F.
Approximate atomic weights: O, 16; H, 1; C, 12; N, 14.
Average composition of air, 23 per cent. of oxygen by weight or 21 per cent. by volume, remainder almost entirely nitrogen.

MOLECULAR WEIGHTS OF GASES

Gas	Formula	Molecular Weight
Carbon dioxide	CO_2	44·00
Carbon monoxide	CO	28·00
Ethylene	C_2H_4	28·03
Methane	CH_4	16·03
Oxygen	O_2	32·00
Water vapour	H_2O	18·02
Hydrogen	H_2	2·016
Nitrogen	N_2	28·02

CHAPTER I

Elementary

History of Internal Combustion Engine—Use of Compression—Comparative Economy.

1. Heat Engines.—Heat engines are machines which receive heat and turn some portion of it into mechanical work. They are of two kinds—

(1) Steam Engines
(2) Internal Combustion Engines.

In steam engines the heat is applied to the boiler which generates steam. The steam passes through the steam pipe into the engine, and when it gets there it makes the engine do work.

The internal combustion engine works in a different way altogether. The heat is actually produced by combustion of fuel *inside* the cylinder of the engine. Whereas the steam engine illustrates external combustion. Gas, oil and petrol engines are called internal combustion engines. A gun is also a form of internal combustion engine in which a certain part of the heat given out on explosion is converted into kinetic energy in the projectile.

2. History of the Internal Combustion Engine.—The first internal combustion engine was made by Huyghens in the year 1680. It was very different to any engine made now. It did not work on gas or oil or petrol, but used gunpowder as its fuel. Gunpowder was exploded in a hollow cylindrical vessel while the piston was at the top, and the air was driven forcibly out. The partial vacuum so caused, tended to pull the piston

down, and this force could be applied by means of a cord and pulley to raise a weight or to do work by some other suitable mechanism. This engine was not a practical success, nor were any later engines working with gunpowder as fuel. In the year 1820 the first engine that could really be called a "gas engine" was made. Its inventor was the Rev. W. Cecil, who by an ingenious arrangement exploded in the working cylinder hydrogen gas mixed with air, the principle being the same as that of the Huyghens gunpowder engine. This also was not a practical success, though a step forward.

3. Lenoir Engine.—The first engines that could be put to practical use were made by Lenoir in 1860. He used the ideas of many people who had previously been working at the subject, but he grasped the matter more thoroughly and much more constructively. Many hundreds of his engines were made. His plan was to draw into the cylinder a mixed charge of air and gas. This he did by causing the connecting rod to pull the piston upwards, then when the piston was half way up the cylinder he ignited, by an electric spark, the gaseous mixture which had been drawn in. The gas could not escape by the inlet pipe, as that contained a non-return valve. So the full effect of the explosion was felt by the piston, which was forced upwards causing the connecting rod to do work. When the piston reached the top of its stroke the outlet valve was opened. Then since the connecting rod was fastened to a crank which turned a fly-wheel, the energy of the latter made the piston descend and drive the burnt products out of the cylinder. It then began to rise again, drawing in a fresh charge of gas and air, which was in turn ignited when the piston was half way up the cylinder. Thus the motion was repeated. The air and gas were always drawn in at about atmospheric pressure, and were ignited by the spark at that pressure. In modern engines the mixture of gas and air is always *compressed* before ignition. Lenoir's engine is therefore called a "non-compression" engine. It used about seven times as much fuel per horse-power as a modern engine.

4. Otto and Langen.—In 1866, Otto and Langen produced an engine in which the piston was not fastened to a connecting rod, but was loose and could fly upwards. When the ex-

ploded gases had expanded and had got cool (partly by expansion and partly by the effect of the cold cylinder walls), the flying piston stopped rising and began to fall under gravity. It was then caught by a kind of ratchet on its connecting rod, and, using its weight in this way, the piston did work. This engine was not very successful, its action being spasmodic. But in 1876 Otto produced what he called a "silent engine" —to distinguish it from the noisy flying piston engine just described. It worked on a principle of operation which had been very clearly stated fourteen years before by Beau de Rochas, who, although he had patented it, had not made a working success of the invention. Otto was a more practical man, and he made his new engine very successful. The method of working was as follows :—

(1) Air and gas were drawn in during an outward stroke of the piston, followed by
(2) Compression of the mixture during the return inward stroke.
(3) Ignition at the inner dead centre, and expansion throughout the next outward stroke.

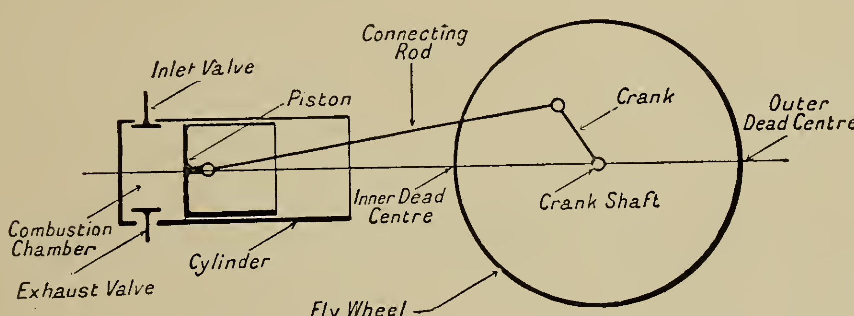

FIG. 1.—Diagram of Otto Engine.

(4) Discharge of the burnt gases on the return of the piston during the last stroke.

This cycle of operations takes, it will be noticed, *four strokes* to complete, and is known as an "Otto-cycle" or "Four-stroke" cycle.

5. Use of Compression.—The fundamental difference between the Lenoir and the Otto engines lies in the fact that the former was a non-compression engine, whilst the latter employed

compression. A further difference is that the Lenoir engine completed its cycle of operations in two strokes, and is known as a "two-stroke" engine, whilst the Otto engine is a "four-stroke" one. The advantage of compression is that the gases are at a fairly high pressure before the ignition point is reached, and so the effect of the explosion is to cause the mixture to reach a far higher pressure, and therefore to do more work, than if the pressure before explosion were no higher than that of the atmosphere outside the engine. The fact that all engines work on compression means that all must have a space provided into which the piston can compress the charge. This space is called the "clearance space." In Fig. 2 the face of the piston, at the end of compression, comes up to the line AB, and on the expansion stroke moves out as far as EF. Then the space between AB and CD is called the "clearance"; the distance from AB to EF the "stroke"; and the ratio of the volumes CDFE to CDBA the "compression ratio," denoted by the letter r. It is obvious that the higher the compression ratio the higher will be the pressure at the end of compression, and thus the higher will be the temperature at that point.

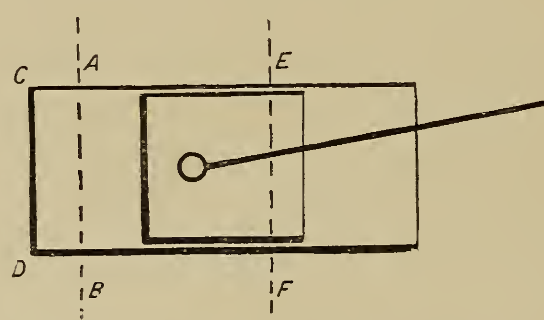

FIG. 2.—Diagram of Cylinder and Piston, showing clearance.

$$\text{Compression ratio} = \frac{\text{volume } C\ E\ F\ D}{\text{volume } C\ A\ B\ D}.$$

6. Dugald Clerk.—In 1880 Dugald Clerk invented an engine which partook of the nature of the Lenoir in that it was a two-stroke engine, and of the nature of the Otto in that the mixture was compressed before explosion. This he did by mixing and slightly compressing the gas and air in a *separate* cylinder instead of in the working cylinder. The working

cylinder received its mixture of gas and air under slight compression, forcing out the exhaust gas as it entered, so that the exhaust and drawing in strokes could be omitted in the main cylinder. Of course the operations thus omitted in the main cylinder had to be done in the other cylinder—called the pump cylinder—but the working cylinder was able to effect twice as many working strokes per minute as before. The work done by this cylinder was therefore doubled, but against this must be set the work lost in pumping in the other cylinder. Engines working on the Clerk cycle are now made in considerable numbers.

7. Daimler.—In 1895 Daimler brought out his well-known high speed petrol engine for automobiles. The Otto cycle was followed, and the chief improvements were of a mechanical nature. Petrol vapour was used instead of gas. This engine gave a great impetus to mechanical transport on roads and to the use of motor-boats.

8. Diesel.—In 1897 a novel form of oil engine was introduced by Diesel. Instead of a combustible mixture of oil-vapour and air being drawn in on the suction stroke, air only was allowed to enter. This was compressed to a very high pressure on the compression stroke—500 lb. per sq. inch—and was raised by this compression to a high temperature. Then at the inner dead centre a small quantity of oil was injected at an even higher pressure (800 lb. per square inch) by means of compressed air. This oil at once ignited on coming into contact with the air, and forced the piston on its outward stroke. As the stroke continued, more oil was injected until the "cut-off" point was reached, when the gases were allowed to expand and do work in the usual way. The effect of admitting the fuel gradually, instead of all at once, was to get a more even pressure on the piston from the beginning of the stroke until the cut-off point. This made the action of the engine similar in some respects to the action of the steam engine, where the principle of gradual admission also applies. In this engine no electric spark was needed to ignite the mixture, since the temperature of the air at the end of compression was itself high enough to cause combustion to take place.

9. Humphrey Gas Pump.—A new type of engine was introduced by Humphrey in 1909, in which the iron piston was replaced by the flat surface of a vertical water column which under the explosive force of the gaseous mixture was made to oscillate in a series of unequal strokes, and in so doing to cause water to pass from a low level tank to a high level one. The water so pumped could if desired be made to work a water turbine.

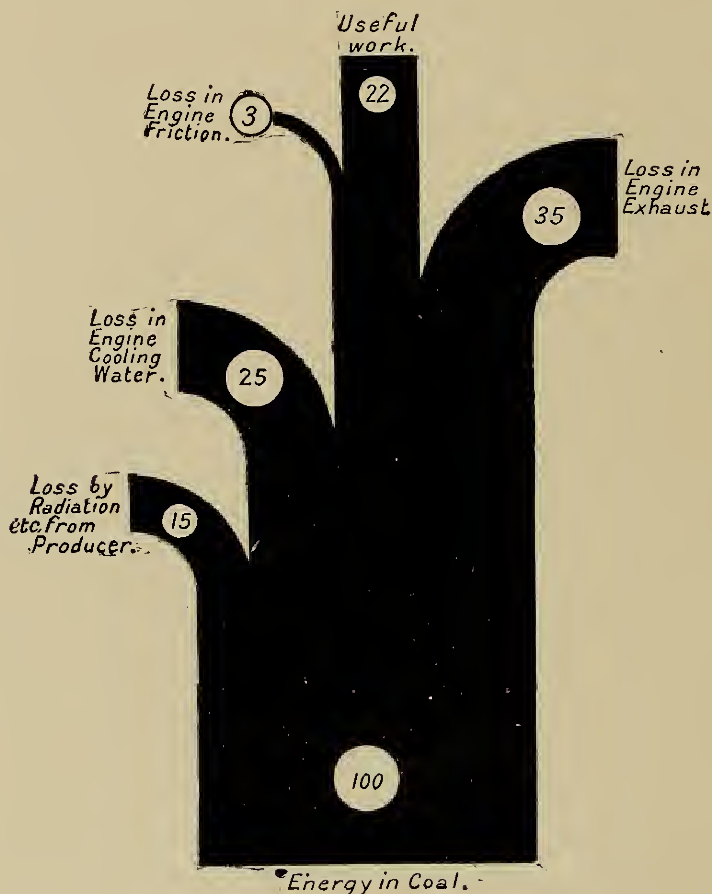

FIG. 3.—Efficiency Diagram of Gas Power Plant.

10. Comparative Economy of Steam, Gas and Oil.—An interesting comparison of the relative fuel consumptions and capital costs was made by T. R. Wollaston, in a paper read before the Society of Chemical Industry in 1914. The following is an extract from his paper:—

For purposes of comparison the following is an approxi-

mate table showing the performance and cost of modern prime movers. The various units are :—

(*a*) The most economical form of modern steam engine in conjunction with modern boiler plant.

(*b*) A steam turbine of highest class with modern boiler plant.

(*c*) Gas engine with "bituminous" gas plant.

(*d*) Diesel oil engine.

Type	B.Th.U. per B.H.P. hour	Fuel cost per B.H.P. hour	Capital cost per B.H.P.
(*a*) Steam engine . . .	19,000	0·102*d*.	£7
(*b*) Steam turbine . . .	21,000	0·112*d*.	£6
(*c*) Gas engine	15,000	0·08*d*.	£8
(*d*) Diesel engine . . .	9,000	0·18*d*.	£8

In the above 70 per cent. efficiency is assumed for boiler and gas plant. Coal of 10,000 B.Th.U. per lb. at 12*s*. per ton. Diesel oil of 18,000 B.Th.U. per lb. at 70*s*. per ton.

In Fig. 3 is seen a diagrammatic representation of the way in which the fuel energy is used in a gas engine and producer plant.

SECTION I

THEORY

CHAPTER II

Thermodynamic Cycles

Units—Perfect Gas—Isothermal Expansion—Adiabatic Expansion—Entropy—Constant Temperature Cycle—Constant Pressure Cycle—Constant Volume Cycle—Air Standard.

11. Quantity of Heat.—The quantity of heat given to the water in a vessel is measured by multiplying the weight of the water in pounds by the rise in temperature of the water due to its receiving this heat. Thus if 40 lb. weight of water be heated so that its temperature rise from 50° C. to 75° C. the quantity of heat supplied is $40 \times 25 = 1{,}000$ heat units, and as the temperature scale used is Centigrade these heat units are called Centigrade heat units (C.H.U.) or pound-calories. The equality of 1,000 pound-calories to 1,800 B.Th.U. can be immediately seen, as one degree Centigrade is equal to 1·8 degrees Fahrenheit. One pound-calorie is the amount of heat required to heat one pound weight of water through one degree Centigrade. One B.Th.U. is the amount of heat required to heat one pound weight of water through one degree Fahrenheit. When the unit of weight is the kilogram and the Centigrade scale of temperature is used, the unit of heat is called the Kilogram-calorie.

12. Specific Heat.—The above definitions have all been expressed in the terms of heating *water*; water being the most convenient standard substance to select, and having, as it happens, a greater capacity for heat than any other known liquid. Thus one pound-calorie will heat one pound of water through one degree Centigrade, but it would heat one pound of mercury through no less than 30 degrees Centigrade. The specific heat of a substance is defined as the quantity of heat

necessary to raise one pound weight of that substance through one degree of temperature. The following are the specific heats of some of the commoner substances :—

Water	1·000
Mercury	0·033
Brass	0·090
Cast Iron	0·12
Lead	0·030
Glass	0·19
Air	0·24

Thus 0·030 pound-calorie are needed to heat one pound of lead through one degree Centigrade, or 0·030 B.Th.U. to heat it through one degree Fahrenheit.

13. Specific Heats of Gases.—In the study of the behaviour of a perfect gas it is usual to assume that the specific heat is independent of the pressure and temperature. In real gases this is found to be only approximately correct, since the specific heat, though practically independent of the pressure, does increase substantially with increase of temperature. This is illustrated in Fig. 21 on p. 63, where the specific heat of the expanding gases in a gas engine is shown plotted against the temperature. When measurement is being made of the specific heat of a gas, it is possible to keep either its volume or its pressure constant. The former specific heat is called the specific heat at constant volume, and is usually denoted by C_v, the latter is the specific heat at constant pressure and is denoted by C_p. The specific heats at constant pressure will always be larger than those at constant volume, because the gas, in expanding, expands against the atmospheric pressure and therefore does work.

The following table gives the constant-pressure specific heat figures at 0° C. for some of the commoner gases :—

Air	0·24
Hydrogen	3·40
Carbon-monoxide	0·24
Carbon-dioxide	0·20

In gas engine work the simplest plan is to take a *mean value* for the specific heat over the temperature range considered.

14. Unit of Work.—As the purpose of internal combustion engines is to turn heat into work it is as important to measure the work done as it is the heat supplied. Work is measured in foot-pounds, one foot-pound being the work done in lifting one pound weight through a vertical height of one foot. If the point of application of a force of P pounds moves through a distance of h feet in the direction of action of the force, the work done is $P \times h$ ft.-lb.

15. Volumetric Heat.—Hitherto specific heat has been defined as the number of thermal units required to raise one pound weight of the substance through one degree of temperature; it is sometimes convenient in the case of gases to know the number of *foot pounds* necessary to raise through 1° C. a mass of gas which at 0° C. and at normal atmospheric pressure (760 mm. of mercury) would occupy a volume of exactly *one cubic foot.** This number is called the "volumetric heat" of the gas to distinguish it from the other way of reckoning. Thus it comes to exactly the same thing whether nitrogen is said to have a "specific heat" of 0·250 or a "volumetric heat" of 27·2 ft.-lb. per cubic foot. To convert specific heat into volumetric heat it is necessary to multiply by the weight in pounds of one cubic foot of the gas (at normal temperature and pressure) and by Joule's equivalent.† It has been found by experiment that for most gases the product of specific heat and density is a constant. The effect of this is that however much the specific heat figures may differ, the figures for "volumetric heat" are almost the same for all gases. This is a great convenience, as it shows that the amount of heat necessary to heat a cylinder full of gases, at a moderate temperature, through any small temperature range, will be about the same whatever the composition of the gases may be. This consideration is of assistance when studying the effect of the presence in an explosive charge of a residuum of burnt gases from the last explosion, particularly when the exact proportion of the different substances in the exhaust products is unknown.

* The letters N.T.P. are often added to show that the volume is measured at Normal Temperature and Pressure.

† *See* p. 14.

16. Efficiency.—The ratio of the energy got out from a machine to the energy put into it is called the *efficiency* of the machine. Thus

$$\text{Efficiency} = \frac{\text{energy given out}}{\text{energy supplied}}.$$

A gas engine usually gives out at the crank-shaft about four-fifths of the energy given to the piston; its efficiency is therefore said to be 0·80 or 80 per cent. This is called the *mechanical* efficiency, to distinguish it from the *thermal* efficiency, which is the ratio of the energy given to the piston to the energy contained in the fuel used, and rarely exceeds 30 per cent. in any engine.

17. Unit of Power.—When a machine is capable of doing 33,000 ft.-lb. of work every *minute* (or 550 ft.-lb. every second), it is said to be of one horse-power, or 1 H.P.

As the work done in a minute by 1 H.P. is 33,000 ft.-lb., so the work done in an hour is $60 \times 33{,}000$ ft.-lb., or 1,980,000 ft.-lb., which is therefore the equivalent of 1 H.P.-hour. Another unit of power, derived from electrical practice, is the kilowatt (or kW.) It is larger than the horse-power and

$$1 \text{ H.P.} = 0{\cdot}746 \text{ kW.}$$

18. Mechanical Equivalent of Heat.—It was at one time thought that when a heat engine did work it did it by passing the heat without loss through the given temperature range, just as the work done by a waterfall depends upon passing a certain amount of water through a certain range of "head." It is now well known that the quantity of heat supplied to an engine is *greater* than that which comes away from it, and that the missing part is the amount of heat that has been converted into mechanical work. Joule was the first to realize that heat could be converted into work and to measure the number of foot-pounds into which one heat unit could be converted. This he did by churning water with paddles so as to produce internal friction in the water. He measured the work done and the rise in temperature of the water; by this means the mechanical equivalent of 1 B.Th.U. was determined. Many

later and more accurate experiments have been made, and the result generally accepted now is that

$$1 \text{ B.Th.U.} = 778 \text{ ft.-lb.}$$
$$\text{and } 1 \text{ pound-calorie} = 1{,}400 \text{ ft.-lb.}$$

19. Changes of State in a Gas or Vapour.—The state of a gas or vapour may be altered by giving heat to it, or by taking heat from it. The state may also be altered by compression or expansion. Any of these processes will bring about changes in one or more of such properties as—volume, pressure, temperature, internal energy, specific heat. These properties are related to one another in various ways, and the two most important of the relationships are called Boyle's Law and Charles' Law.

20. Boyle's Law.—Boyle's Law states that if the temperature be kept constant the volume of a mass of gas will vary inversely as the pressure.

In symbols—

$$PV = \text{Constant}$$
(for constant temperature).

21. Charles' Law.—Charles' Law states that if the pressure be kept constant, equal volumes of different gases increase equally for the same increase in the temperature ; also, that if a gas be heated under constant pressure equal increments in its volume correspond very closely to equal intervals of temperature.

22. Absolute Temperature.—It is found by experiment that the amount by which the volume of gas expands when its temperature is changed by one degree Centigrade (the pressure being constant) is $\frac{1}{273}$rd part of its volume at 0° C. If this proportion held rigorously for all temperatures, however low, it would follow that at a temperature of 273 degrees *below* 0°C. the volume of the gas would be zero. The temperature of — 273° C. is therefore called the *Absolute Zero*, and temperatures reckoned from this zero instead of 0° C. are called *absolute temperatures*. Thus the boiling point of water is called either 100° C. or 373° C. (absolute). When using the Fahrenheit scale the number 459 should be added to the ordinary Fahrenheit temperatures to bring them to Fahrenheit tem-

peratures (absolute). Charles' Law may therefore be expressed—

If V=volume at θ° C,
and V_0=volume at 0° C.

$$V=V_0+\frac{V_0}{273}\theta$$
$$=\frac{V_0}{273}(\theta+273)$$
$$=\frac{V_0}{273}T,\text{ where } T=\text{absolute temperature,}$$

i.e., $\frac{V}{T}$ = constant (for constant pressure).

23. Perfect Gas.—A perfect gas is defined as one which satisfies rigorously both these laws, which may be combined into $\frac{PV}{T}$ = constant. This constant is usually written R; thus $PV = RT$. Most of the ordinary gases comply very closely with the perfect gas laws, particularly at the temperatures met with in internal combustion engines. The equation $\frac{PV}{T}$ = constant applies to *any* weight of gas; when a standard weight of gas (e.g. 1 lb.) is considered, then the value of R depends on the nature of the gas. For 1 lb. weight of air $R = 96$, the units being pounds, feet and degrees Centigrade.

24. To prove when unit weight of gas is considered that

$$PV=RT=J(C_p-C_v)T$$

where R is a constant and J is the value of the mechanical equivalent of heat ("Joule's Equivalent").

Consider *one pound weight* of gas (at P_0, V_0, T_0) confined in a cylinder of exactly one square foot in cross-sectional area and having above it a piston whose weight may be neglected. Let the temperature increase to T_1 and the volume to V_1, keeping the pressure constant and equal to P_0.

The heat supplied to the gas =

$$C_p(T_1-T_0)\text{ heat units}$$

equivalent to $JC_p(T_1-T_0)$ ft.-lb.

The external work done by the gas$=P_0(V_1-V_0)$ ft.-lb.

which$=R(T_1-T_0)$ ft.-lb.

(using the equation $PV=RT$)

Then the internal energy remaining in the gas must be equal to the difference of those two, or

$$=(JC_p-R)(T_1-T_0) \text{ ft.-lb.}$$

Now Joule discovered experimentally that the gain in internal energy of a gas depends only on the initial and final temperatures, and is independent of changes of pressure or volume, i.e., that the above increase in internal energy is the same as if the temperature had risen while the volume remained constant, in which case the heat units required would have been $C_v(T_1-T_0)$, or in energy units $JC_v(T_1-T_0)$ ft.-lb.

Thus $(JC_p-R)(T_1-T_0)=JC_v(T_1-T_0)$

or $R=J(C_p-C_v)$

This shows that the perfect gas law may be written

$$\frac{PV}{T}=J(C_p-C_v)$$

a form which is often convenient. It shows also that for any gas which obeys the perfect gas law the specific heat at constant pressure is always larger than the specific heat at constant volume by the same amount, no matter what the temperature or pressure may be. So that if one specific heat be known the other, or the ratio of the two, can easily be calculated.

25. The equation

$$\frac{PV}{T}=R=J(C_p-C_v)$$

is true for all perfect gases, the quantity present being unit weight. It may be written

$$\frac{P}{T}=\frac{J}{V}(C_p-C_v).$$

If we take two different gases, both obeying the perfect gas law, and adjust their pressures so as to be equal, and also their temperatures to be equal, the two values of $\frac{1}{V}(C_p-C_v)$ must be the same. But the weights of the gases being the same,

the volumes occupied must be inversely proportional to their densities. Thus $(C_p—C_v)\times$ density must be a constant quantity for such gases.

The following table shows how real gases approximate to this—

Gas	C_p	C_v	Density relative to Air	$(C_p—C_v)\times$ density
H_2 . . .	3·409	2·406	0·0692	0·069
N_2 . . .	0·244	0·173	0·970	0·069
O_2 . . .	0·218	0·155	1·105	0·070
CO_2 . . .	0·217	0·171	1·520	0·070

The explanation why there are any differences at all is because these gases are not "perfect gases." The assumption is implied, moreover, that the specific heat is independent of temperature, and although for many calculations this is sufficiently nearly true, there are others, as will appear in a subsequent chapter, in which this is by no means the case.

26. Ratio of Specific Heats.—The ratio of the two specific heats of a gas is an important one, and is generally called by the Greek letter γ, thus

$$\gamma=\frac{C_p}{C_v}$$

Since $J(C_p—C_v)=R$

$$\gamma-1=\frac{C_p}{C_v}-1=\frac{R}{J.C_v}$$

γ is usually from 1·3 to 1·4; and for air is exactly 1·41.

27. Isothermal Expansion.—When a gas expands so that the temperature is always constant the expansion is said to be *Isothermal*.

In symbols—

$$PV=\text{constant}.$$

This is, of course, Boyle's Law.

(This is occasionally referred to as a "hyperbolic" expansion as the graph of the above equation is a hyperbola.)

28. Adiabatic Expansion.—When a gas expands in such a way that heat, as such, is neither given to it nor taken from it,

the expansion is said to be *adiabatic.* Such an expansion, or compression, may be imagined as taking place in a cylinder made of a completely non-conducting material, no heat being generated by chemical action nor lost by radiation. The more quickly an expansion or compression takes place, the more nearly is the adiabatic law followed, since there is a shorter time for any transfer of heat to take place. The rapid heating of a tyre-pump when used vigorously is a familiar phenomenon.

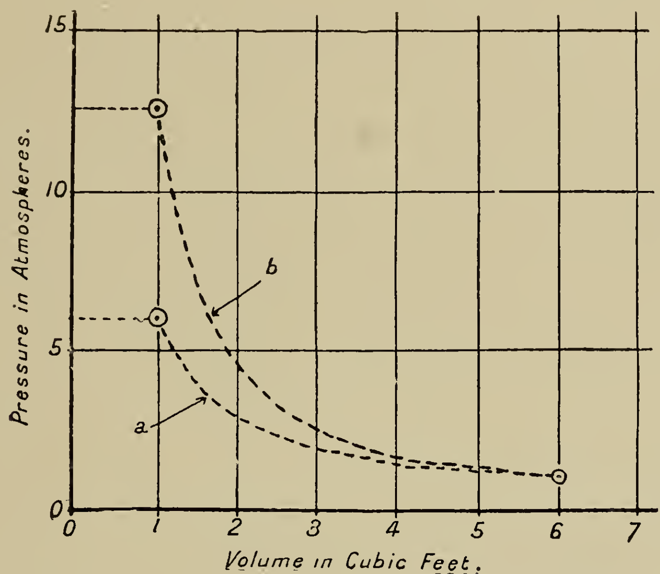

FIG. 4.—*P V* diagram showing compression of six cubic feet of air into one cubic foot. (*a*) Isothermal; (*b*) Adiabatic. [Final pressure is more than twice as high in (*b*) as in (*a*).]

When the expansion is adiabatic the law connecting P and V for a perfect gas can be shown to be

$$PV^{\gamma}=\text{constant}.$$

In Fig. 4 is shown the result of compressing a mass of gas from 6 cu. ft. to 1 cu. ft. Such compressions are approximately adiabatic—see curve *b*—when the process is carried out very rapidly; and approach the isothermal—see curve *a*—when the compression is so slow that most of the heat is dissipated during the time taken by the compression.

29. Proof.—" Joule's Law," quoted in paragraph 24, comes to this, that the gain in internal energy due to rise of tempera-

ture must equal the difference of energy due to heat supplied and the work done.

Thus, if ΔH heat units are supplied to unit weight of gas

$$J.C_v.\,\Delta T = J.\,\Delta H - P.\,\Delta V$$

where ΔT and ΔV are increments in temperature and volume.

If the gas is neither to receive nor to lose heat

$$\Delta H = O$$

and the equation simplifies to

$$J.C_v\,\Delta T + P.\,\Delta V = O.$$

Now in any finite transformation P will be continually changing, and the process must therefore be imagined to be split up into a great number of infinitesimal steps. Consider ΔT and ΔV as infinitesimal increments, and obtain the equation connecting P and V by integration, thus :—

$$J.C_v.\,\Delta T + P.\,\Delta V = O$$

from pars. 24 and 26, $PV = J.C_v.(\gamma - 1)T$

therefore $P.\,\Delta V + V.\,\Delta P = J.C_v(\gamma - 1)\,\Delta T$

$= -(\gamma - 1)P.\,\Delta V$

therefore $V.\,\Delta P = -\gamma\,.P.\,\Delta V$

$$\frac{\Delta P}{P} + \gamma\frac{\Delta V}{V} = 0$$

in the limiting case $\dfrac{dP}{P} + \gamma\dfrac{dV}{V} = 0,$

therefore $\log P + \gamma \log V = \text{constant}$, or $PV^{\gamma} = \text{constant}$.

30. Temperature Changes in Adiabatic Transformations.—The adiabatic law for a perfect gas is

$$PV^{\gamma} = \text{constant}\,;$$

combine this with the perfect gas equation of

$$PV = RT$$

and eliminate P

then $TV^{\gamma-1} = \text{constant}\,;$

or V can be eliminated and then

$$\frac{T^{\gamma}}{P^{\gamma-1}} = \text{constant}.$$

If, therefore, the initial state of a mass of gas be known it

is possible to calculate its temperature at any point after adiabatic expansion or compression from a knowledge either of its new volume or of its new pressure.

Both the laws discussed in paragraphs 27 and 28 are special cases of the general formula $PV^n = \text{constant}$, n being equal to unity in the isothermal case and equal to γ in the adiabatic case. In the internal combustion engine the gas does not expand or compress according to either of these laws precisely, but the expansions and compressions do in every case follow very nearly *some* law of the type $PV^n = \text{constant}$, where n has a value lying between unity and γ.

Example.—If the gas during a compression stroke increased in pressure from atmospheric pressure to 65 lb. per sq. inch above the atmosphere, and if the temperature before compression were 120° C., the temperature at the end of compression could be calculated from the equation in par. 30.

$$\frac{T^n}{P^{n-1}} = \text{constant}$$

and if n be 1·3

$$\text{Then } \frac{T^{1\cdot3}}{(14\cdot7 + 65)^{0\cdot3}} = \frac{(120 + 273)^{1\cdot3}}{(14\cdot7)^{0\cdot3}}.$$

$$\text{or } T = (120+273)\left(\frac{79\cdot7}{14\cdot7}\right)^{\frac{0\cdot3}{1\cdot3}} = 393\left(\frac{79\cdot7}{14\cdot7}\right)^{0\cdot23}$$

or $T = 580°$ C. (absolute) $= 307°$ C.

This explains how it is that a gas gets hot when compressed so suddenly that there is little time for heat to escape through the walls of the cylinder.

31. The Thermodynamic Laws.—The following are the two fundamental laws of thermodynamics.

(1) In all transformations, the energy due to the heat units supplied must be balanced by the external work done plus the gain in internal energy due to the rise in temperature.

(2) It is impossible for an automatic machine, unaided by any external power, to convey heat from a colder to a hotter body.

The first of these laws was discovered experimentally by Joule. It has been stated in paragraph 29, and was there interpreted in symbols, viz. :

$$J.\,\Delta H = JC_v\,\Delta T + P\,\Delta V$$

The second law may be said to represent universal experience in the working of heat engines.

32. Thermal Efficiency.—So far as the first law is concerned there is nothing to show why *all* the heat supplied to an engine should not be converted into work. But the effect of the second law is that only a portion of the heat supplied can be converted into work, and, as stated in par. 16, the ratio

$$\frac{\text{Heat converted into work}}{\text{Heat supplied to engine}}$$

is known as the thermal *efficiency* of the engine. The better the engine the higher the efficiency. The most efficient heat engine yet built has an efficiency of about 0·4.

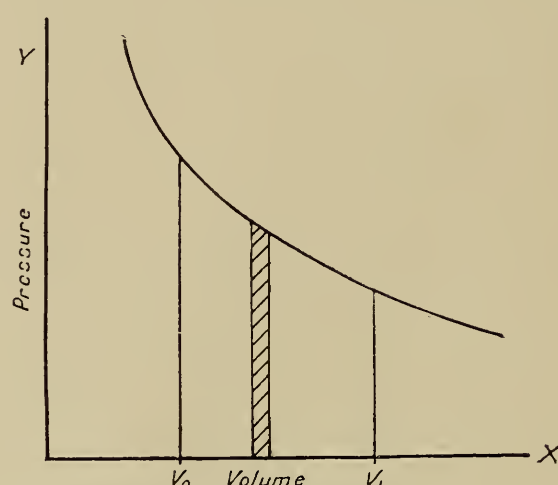

FIG. 5.—PV diagram.

33. Application of Graphical Methods to Thermo-dynamics; Pressure-Volume and Temperature-Entropy Diagrams.—The reader is probably familiar with graphical methods as applied to physical problems. In many such cases it is customary to deal with three physical quantities; two of these are plotted along the axes of co-ordinates, and the relation between them exhibited by the graph, while the third is involved in the area contained between the curve and one of the axes.

If pressure (lb. per sq. ft.) be plotted along one axis and volume (cu. ft.) along the other, as shown in Fig. 5, the area between the curve and the X axis, bounded by the ordinates

at $V = V_0$ and $V = V_1$, will give the external work done (ft.-lb.) when the volume of the gas increases from V_0 to V_1.

Proof.—Area of shaded strip $= P.\Delta V$, which is work done by pressure in increasing volume by ΔV. Therefore total work done $= \Sigma P.\Delta V$ for all such strips (or in calculus notation, $\int P.dV$) which is the area under the curve.

It is this principle that enables the work done by an engine to be calculated from an indicator diagram showing the pressures and volumes of the working medium.

In some problems, however, it is convenient to have the *temperature* shown along the Y axis, and the area under the

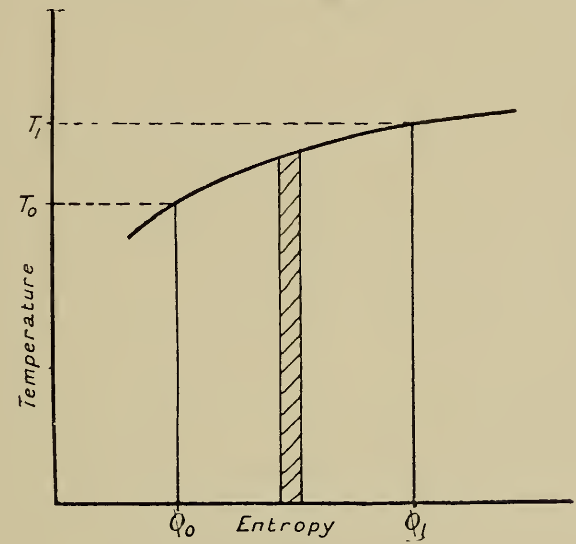

FIG. 6.—$T\phi$ diagram.

curve to show, not work done, but heat units supplied to the gas. The question arises, what, in this case, is to be plotted along the X axis? The answer to this question is that the quantity to be plotted along the X axis is not one like pressure and volume with which acquaintance has already been made, but a new one, and one which cannot be measured directly. The name given to it is *entropy*. It is not possible to give a simple scientific definition of entropy, nor is it necessary to do so. It is obviously some property of the state of a gas which determines the connexion between rise of temperature and increase of heat units. If we keep in mind the graphical in-

terpretation as explained above, it is unnecessary to express the idea of entropy in any formal definition.* In Fig. 6 such a graph is given. The area under the curve, and lying between the ordinates at φ_0 and φ_1, measures the number of heat units supplied to the gas between the temperatures T_0 and T_1.

Calculation.—Calling the entropy φ, the area of the shaded strip $= T . \Delta \varphi$, but this by definition is equal to ΔH,

$$\text{therefore } T . \Delta \varphi = \Delta H$$

$$\text{or } \Delta \varphi = \frac{\Delta H}{T}$$

$$\text{and } \varphi = \int \frac{dH}{T}$$

From this formula the actual value of the entropy in a mass of gas can be calculated.

34. Unit of Entropy.—If the area under the curve in Fig. 6 were 1000 pound-calories and the temperature had remained constant at 500° C. (absolute), corresponding to an isothermal expansion, the curve would have been flat, i.e. a straight line parallel to the axis of entropy, and it is clear that the difference $(\varphi_1 - \varphi_2)$ would have had to be two units of entropy in length, so that

$$2 \times 500 = 1000 \text{ pound-calories.}$$

One unit of entropy would therefore be the amount of increase in entropy due to the reception of a number of heat units equal in amount to the absolute temperature at which the heat is received, and this unit of entropy is called 1 *rank*.

The temperature values used in entropy calculations must *always* be absolute. The importance of temperature-entropy graphs lies chiefly in their applications to isothermal and adiabatic transformations :—

(1) In isothermal transformations the temperature is constant, so that the graph will be a straight line parallel to the entropy axis.

(2) In adiabatic transformations no heat units are gained or lost, so that the entropy remains constant and the graph will be a straight line parallel to the temperature axis.

* Readers desiring to get a fuller idea of entropy are referred to Professor Callendar's address to the Physical Society, of which an abstract is given on p. 97 of *Nature* for March 16, 1911.

This means that any closed circuit made up of successive isothermal and adiabatic compressions and expansions will have a graph composed exclusively of straight lines at right angles to one another. Hence the area can be very easily measured, and the amount of heat supplied be readily determined.

35. PV and T φ Diagrams Compared.—The following statements help in memorizing the relationships between these two:—

(1) *Average force* (*lb.*) $\times$ *space range* (*ft.*) = *work done* (*ft.-lb.*), or, what comes to the same thing, *average pressure* (*lb. per sq. ft.*) $\times$ *volume range* (*cu. ft.*) = *work done* (*ft.-lb.*).

(2) *Average temperature* (*absolute*) $\times$ *entropy range* (*ranks*) = *heat units*,—the latter being either calories or B.Th .U.

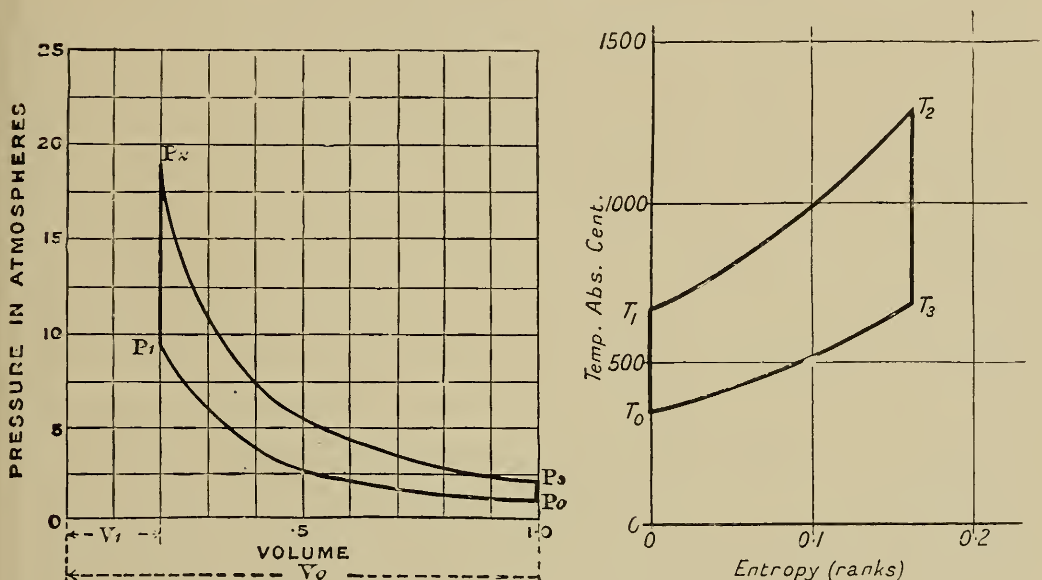

FIG. 7.— PV and Tϕ diagrams for Constant-volume Cycle.

according as the temperature (absolute) has been measured in the Centigrade or Fahrenheit scales.

36. Areas of Closed Cycles.—After any ideal cycle of operations, when the gas returns to its initial state, both the PV and T φ diagrams will be closed figures ; in this case the net work done (in the PV diagram) and the net heat units taken (in the T φ diagram) will be given by the area of these closed figures.

Thus the PV and T φ curves shown in Fig. 7 are those of the

"Otto" cycle on which most modern gas engines work, and they will be referred to at greater length in this book.

The area under the curve T_1T_2 = heat units received.

The area under the curve T_3T_0 = heat units rejected.

Thus the area contained within the closed figure $T_0T_1T_2T_3$ gives the number of heat units which are converted into work by the engine. If this be multiplied by the numerical value of J, it will give the same result as would be obtained by measuring the area of the figure $P_0P_1P_2P_3$.

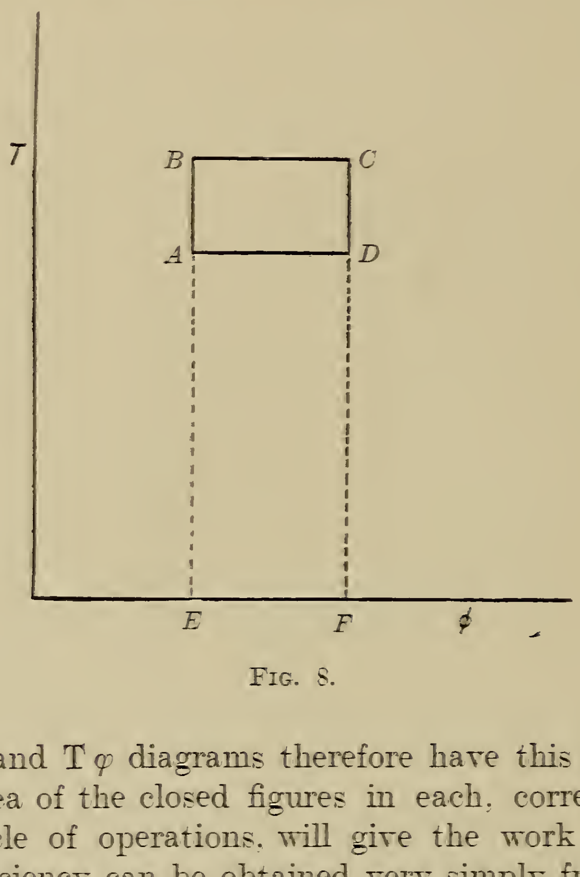

Fig. 8.

The PV and T φ diagrams therefore have this in common, that the area of the closed figures in each, corresponding to a given cycle of operations, will give the work done. The thermal efficiency can be obtained very simply from the T φ diagram since

$$\text{Thermal efficiency} = \frac{\text{area } T_0T_1T_2T_3}{\text{area under } T_1T_2}.$$

In Fig. 8 is shown a very simple entropy diagram for one pound of gas. The gas starts at the point A; the temperature is then increased to the point B, whilst the entropy remains

constant—an adiabatic compression; then the gas has its temperature kept constant from B to C, whilst the gas receives heat and the entropy increases—an isothermal expansion; then from C to D the gas expands adiabatically as the entropy is constant and the temperature falls to D; then from D to A the temperature remains steady, whilst the gas gives up its heat and the entropy diminishes from D to A, so bringing the gas back to its original state, and ready to go through the cycle again. This is the well-known **Carnot Cycle,** which is so often shown on the PV diagram, but is so much more easily understood on the $T\varphi$ diagram.

In this case thermal efficiency $= \dfrac{\text{area ABCD}}{\text{area EBCF}} = \dfrac{AB}{EB}$

$$= \frac{\text{max. temp. of cycle—min. temp. of do.}}{\text{max. temperature of cycle}}$$

which is the customary expression for the efficiency of the Carnot Cycle. This is an instance of how simple the use of the $T\varphi$ diagram makes such calculations.

37. In this last named figure all the lines were parallel to one or other of the axes. This was because an ideal cycle of the simplest nature was being followed. In Fig. 9 the sloping lines AB and BC have been drawn at random. What changes of state would they represent?

The line AB shows an increase of both entropy and temperature, both of them increasing at about an equal rate. So that heat is being given to the gas, and the temperature is increasing meanwhile. This is generally similar to what goes on during explosion in a gas engine cylinder, as the gas takes in heat from the effect of chemical combination, and the temperature rises while it does so. Having arrived at the point B the gas now follows the line BC, during which the gas continues to take in heat, and the temperature decreases. This is what would occur, on a lesser scale, in a gas engine cylinder were the combustion of the gas to continue right through the working stroke instead of ending at the point of highest temperature, as it is now generally believed to do. Then to get the gas back to its original state the line CA is followed, and during it the gas gives out its heat at a nearly

steady temperature, i.e. almost an isothermal compression. No gas engine works exactly on this cycle, which was one drawn at random to show how any cycle whatsoever can be very easily and readily studied by the use of the T φ diagram. It is obvious from the diagram that the efficiency of this triangular cycle would be a low one as the area is small having regard to the temperature variation represented.

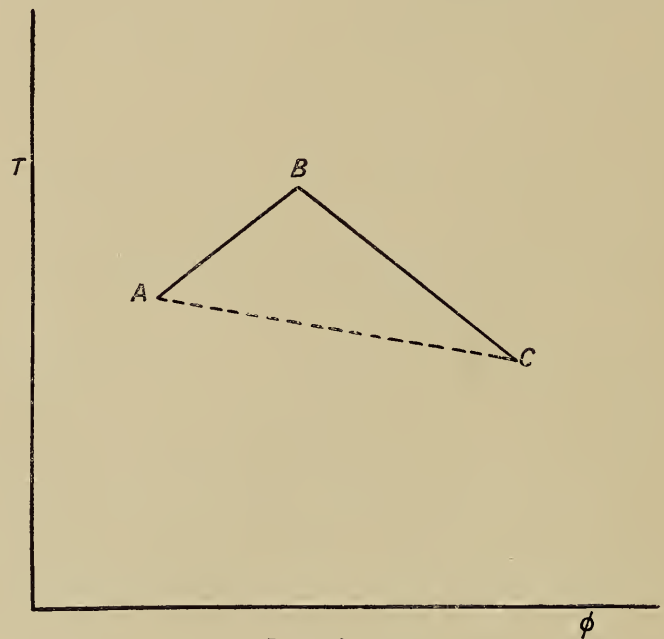

FIG. 9.

Gas engine indicator diagrams are often turned into T φ diagrams, but it is necessary that certain precautions should be taken in doing so. The difficulty lies in the fact that the working fluid does not remain in the cylinder for a number of cycles, but is periodically discharged to exhaust, and a fresh charge brought in. The cycle can, however, be treated as a continuous one if the exhaust gases are considered to have their relatively high temperature and pressure reduced to those of the incoming charge, the volume being kept constant. In an Appendix to an Institution of Civil Engineers report * Captain Sankey has shown a number of PV and T φ diagrams for the same gas engine cycles, and by the permission of

* *I. C. E. Proc.*, Vol. 162.

the Council of the Institution, one of them is reproduced in Fig. 10. The outer lines show the T φ and PV diagrams for

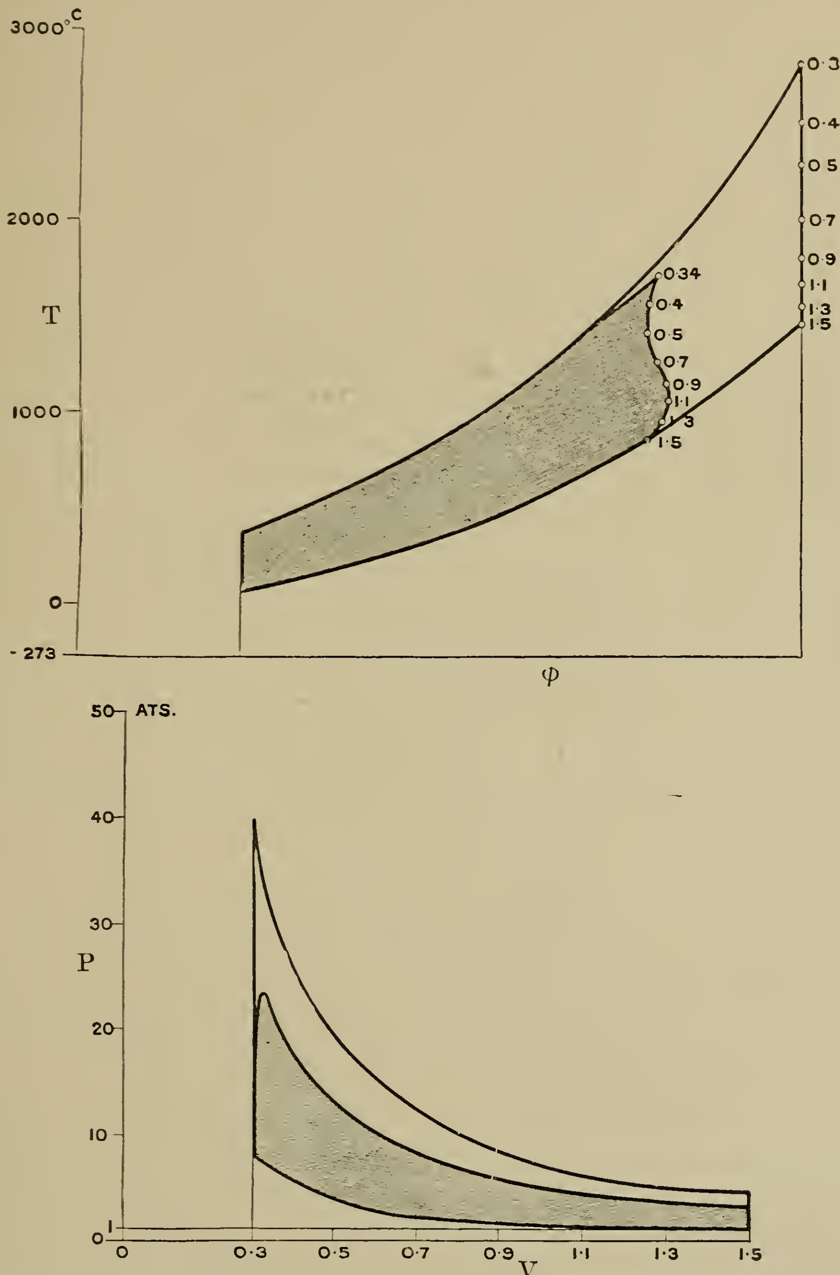

FIG. 10.

an ideal engine, whilst in the shaded portion is given the same diagrams for a probable *actual* engine. The wavy part of the entropy curve shows the expansion period of the cycle.

It has been drawn to show the loss of heat to the walls and piston during the beginning of expansion, and the subsequent flow of heat in the reverse direction during the latter part of the stroke,* this effect dying away again at the very end of the stroke, possibly on account of the slow motion of the piston at that point, which would allow the walls a greater amount of time in which to part with their heat.

Before dealing with the efficiencies of the various cycles of working it is necessary to say something about the working medium. The gaseous mixture that enters a gas engine (for oil or petrol engines the same considerations apply) is usually $\frac{9}{10}$ air and the rest gas, and even when the proportion of air is not quite so high as this, by far the greater part of the mixture is simply air. Air is in fact the working substance, and gases, oils and petrols are used merely to raise its temperature to the point required to carry out the predetermined cycle of operations. So that although the thermal constants are given not only for air but also for other gases, etc., it must be remembered that air is the most important factor, and that inasmuch as air is $\frac{4}{5}$ nitrogen, it is the latter gas which is most concerned, however passively, in the working of internal combustion engines. The following table shows the composition of the fuel gases chiefly in use, and their approximate calorific values.

	Town Gas	Producer Gas	Blast Furnace Gas	Coke-Oven Gas
	per cent.	per cent.	per cent.	per cent.
CO	7	20	25	8
CO_2	2	9	6	2
H	46	21	2	53
N	3	48	66	5
Hydrocarbons .	42	2	1	32
B.Th.U. per cub. ft. about . .	600	150	90	540

38. Ideal Standard Cycles.—Every one who is acquainted

* It would have been more accurate to have shown a continuous loss of heat by the gas—*see* par. 59.

with steam engines knows that the standards of comparison are the Carnot Cycle and the Rankine Cycle, that is to say, these two ideal cycles of operation are the standards by which actual engines are best judged. It would be unfair to complain of any engine which gave a thermal efficiency of 0·27 when that ideally possible for the temperatures employed was only 0·30, indeed such an engine must be greatly superior to any yet constructed, and although 27 per cent. efficiency does, it is true, mean that 73 per cent. of the energy is wasted, yet in reality the engine is a very good one as it yields $\frac{\cdot 27}{\cdot 30}$, i.e. 90 per cent. of what is ideally possible. It is this figure of 90 per cent. which should really be looked to. The figure of 0·27 gives little information, but the figure of 90 per cent. shows at once that unless the manner of working be altogether changed there is only 10 per cent. left to improve upon. In a steam engine the endeavour is to keep the cylinder hot and so prevent the condensation which causes the efficiency to fall below its possible level. In a gas engine, on the contrary, the endeavour is to cool the cylinder to keep the engine from jamming and otherwise working badly. Clearly there is here a marked difference in operation, and correspondingly it becomes necessary to devise new standards of comparison suitable to the working of gas engines.

There are **Three Ideal Standard Cycles,** viz.—

1. The constant temperature type.
2. The constant pressure type.
3. The constant volume type.

Each of these has been investigated by a Committee appointed by the Institution of Civil Engineers, and as it is desirable to avoid a multiplicity of methods of dealing with the same thing, the author will follow generally the procedure they recommend.

39. The Constant Temperature Type.—In an engine of this type, all the heat is taken in at the highest *temperature* and *all* is afterwards rejected at the lowest *temperature*. This is what has been defined above as the Carnot Cycle, and it can be proved that for the same temperature limits no possible

treatment of a heat engine can give a higher efficiency than is theoretically obtainable in this way. The diagrams in Fig. 11 show at once that the efficiency is $\frac{T_1 - T_0}{T_1}$ where T_1 is the highest temperature and T_0 the lowest, both of course being reckoned from the absolute zero of temperature. T is always used in this book to mean temperature absolute, and θ to mean temperature as read on a thermometer. A PV diagram is also shown and any one acquainted with the working of steam or gas engines would recognize that for any given h.p. the cylinder would require to be exceedingly large and

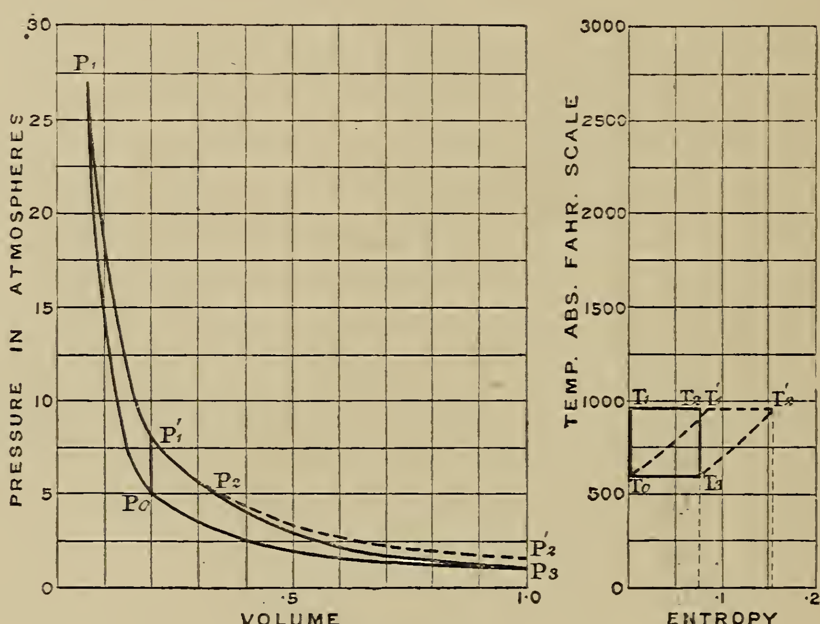

FIG. 11.

costly, so that the extra economy in the matter of fuel brought about by its high efficiency would be more than counterbalanced by the inconvenience of the size of the engine and by the extra annual outlay necessary to provide for interest and depreciation on the enhanced capital cost.

No gas engine works on this cycle or indeed on anything very like it. It is not, therefore, quoted nearly so often in gas engine work as in steam engine practice.

40. The Constant Pressure Type.—In this type of engine all the heat is received at the highest *pressure* and rejected at the lowest *pressure*.

T φ and PV diagrams are shown for this cycle in Fig. 12.

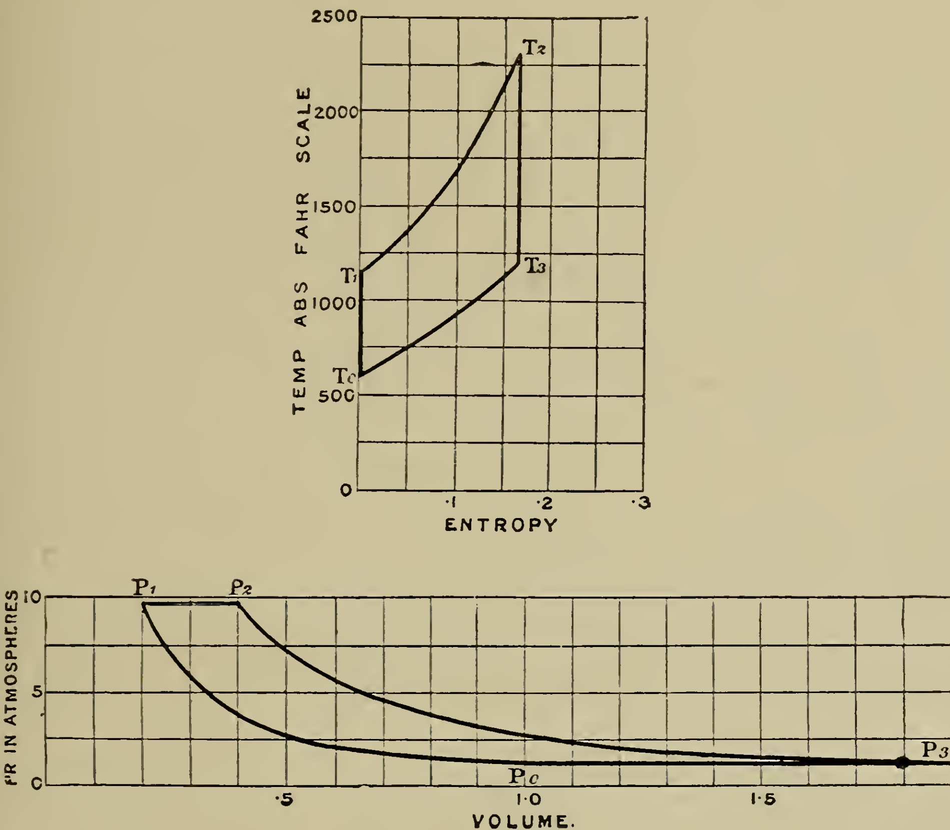

FIG. 12.

The heat received per lb. of gas in this case is $(T_2 - T_1) \times C_p$, and that rejected is $(T_3 - T_0) \times C_p$, so that

$$\text{thermal efficiency} = \frac{\text{heat taken in} - \text{heat rejected}}{\text{heat taken in}}$$

$$= 1 - \frac{T_3 - T_0}{T_2 - T_1}.$$

During the parts of the cycle shown by the lines T_0T_1 and T_2T_3, heat is being neither received nor rejected by the gas; the expansion and compression must therefore be adiabatic.

For adiabatic expansions, $PV^{\gamma}=$ constant, and by par. 30

$$\frac{T^{\gamma}}{P^{\gamma-1}}=\text{constant.}$$

$$\text{Therefore } \frac{T_0}{T_1}=\left(\frac{P_0}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ and } \frac{T_3}{T_2}=\left(\frac{P_0}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\text{Thus } \frac{T_0}{T_1}=\frac{T_3}{T_2}=\frac{T_3-T_0}{T_2-T_1}=\left(\frac{P_0}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\text{Therefore } \eta=1-\left(\frac{P_0}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\text{The compression ratio } r=\frac{V_0}{V_1}=\left(\frac{P_1}{P_0}\right)^{\frac{1}{\gamma}}$$

$$\text{Therefore } \eta=1-\left(\frac{1}{r^{\gamma}}\right)^{\frac{\gamma-1}{\gamma}}$$

$$=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

This gives the value of the efficiency of this cycle in terms of r the compression ratio. It is an important fact that this efficiency is independent of the temperatures and pressures attained, and depends only on the ratio of compression. It shows that for high efficiencies the compression must be high.

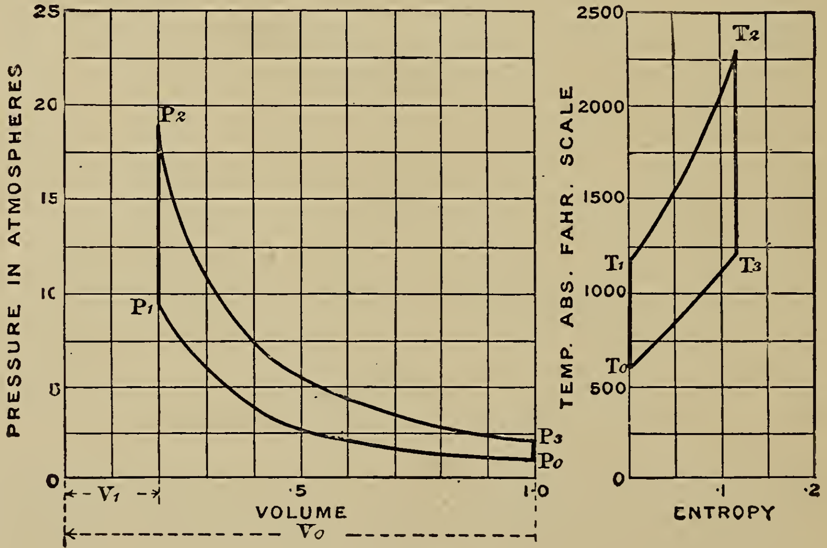

FIG. 13.

The Brayton and Diesel engines approach most nearly to this cycle.

41. The Constant Volume Type.—In this type all the heat is received at constant volume and rejected also at constant volume. These two volumes are the volume at ignition and the volume at exhaust. This cycle may also be called the Otto or Beau de Rochas Cycle, and it is the one on which practically all modern gas engines work or attempt to work. The diagrams in Fig. 13 show the working of the cycle.

The efficiency is calculated in the same manner as the previous one; heat taken in $=(T_2-T_1)C_v$ and heat rejected $=(T_3-T_0)C_v$, from which it follows that

$$\text{Efficiency}=\eta=\frac{(T_2-T_1)C_v-(T_3-T_0)C_v}{(T_2-T_1)C_v}$$

$$\therefore\ \eta=\frac{T_2-T_1-T_3+T_0}{T_2-T_1}$$

$$=1-\frac{T_3-T_0}{T_2-T_1}$$

Then as before

$$\frac{T_1}{T_0}=\frac{T_2}{T_3}=\left(\frac{V_0}{V_1}\right)^{\gamma-1}=r^{\gamma-1}$$

Therefore
$$\eta=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

And this it will be noted is exactly the same expression as before. Indeed, the Carnot Cycle can also have its efficiency expressed in exactly the same way, but it must be remembered that although the efficiency of all three cycles depends upon the degree of compression and would be the same in all were the compression ratios the same, yet the temperature ranges would be very different, and it would be found that the Carnot Cycle gave the least range of temperature for any given efficiency. The discovery that for the same compression ratios the same efficiency holds good for each of these three cycles is attributed to Professors Unwin and Callendar.

In view of the simplicity of this result it is not difficult to understand that the Committee of the Institution of Civil

Engineers, appointed to inquire into the matter, should have selected for use as the best expression for the ideal efficiency the form—

$$\eta=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

This expression therefore holds the place in gas engine work which in the steam engine is filled by the well-known

$$\frac{T_1-T_0}{T_1}.$$

42. The remaining point to be considered is the value to give to γ. The gaseous mixture which works in gas engines depends upon whether lighting gas, producer gas, blast furnace gas or coke-oven gas is being employed, and with oil and petrol engines other mixtures occur. It is evidently impossible therefore to get a value for γ which will accurately suit all engines. It must be remembered, however, that the working fluid always consists chiefly of air, and it has been urged by some engineers that, having regard to the preponderance of the atmospheric oxygen and nitrogen in all internal combustion engines, little error could arise if it were all assumed to be air. The "**Air Standard**" for efficiency resulted. It assumes that air is the working fluid (and that the small quantity of combustible gas is merely used to heat this air by combustion), and that γ has the air value of 1·40, so that

$$\eta=1-\left(\frac{1}{r}\right)^{0\cdot4}$$

This expression gives for different values of r the following theoretical efficiencies—

r	η
2	0·242
3	0·356
4	0·426
5	0·475
7	0·541
10	0·602
20	0·698
100	0·841

In practice 50 to 60 per cent. of these efficiencies are usually obtained, and it is clear that a comparison between different engines can be made by noting what percentage of the ideal efficiency is obtained, in each case, for the compression ratio at which each works. A natural result of this rise of efficiency

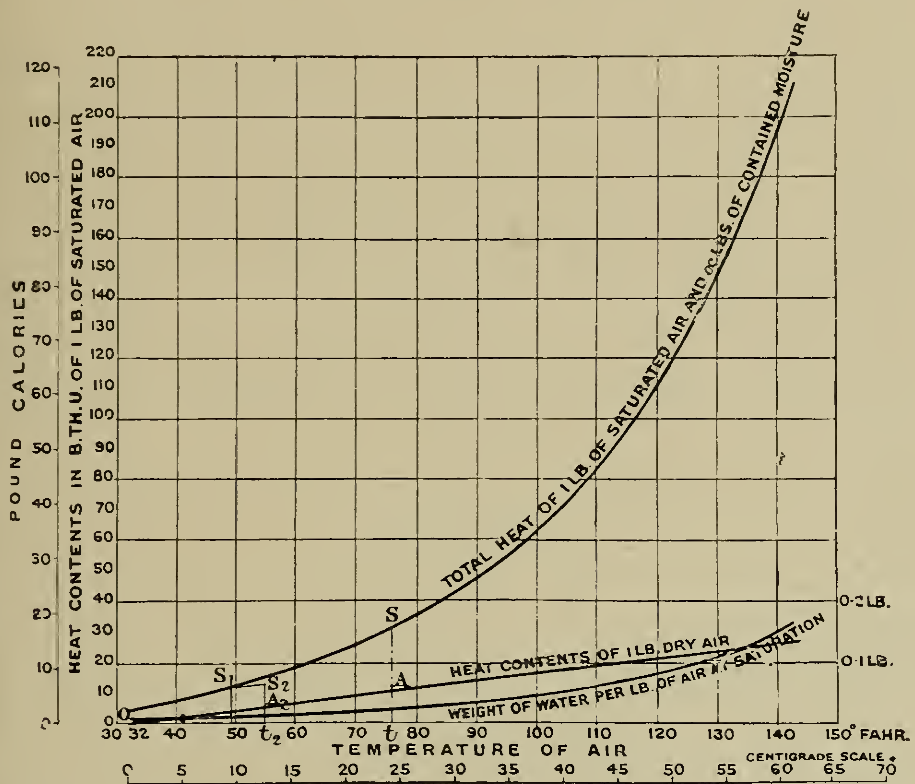

FIG. 14.—Curves showing heat contained in 1 lb. of saturated air at various temperatures. Thus tS represents the heat content of 1 lb. of dry air and the associated quantity of water vapour, which occupies the same volume as the 1 lb. of air at temperature t.

with compression is that taken over a long range of years there has been a decided increase in compression pressures. It is indeed this movement which is the chief cause of the great advances that have been made in the heat economy of gas engines. Thus in 1880 a compression pressure of 30 or 40 lb. per sq. inch was usual. Now the compression pressure sometimes goes up to 170 lb. per sq. inch when working with producer gas and with the Diesel oil engine as high as 500 lb. per sq. inch. The effect of high compression pressures is illus-

trated in practice by the following comparative figures. It was found that an engine * working with a compression pressure of 120 lb. used 11,500 B.Th.U. per B.H.P.-hour, whereas one working with a corresponding pressure of 170 lb. used only 9,500 B.Th.U.

43. The Council of the Institution of Civil Engineers have permitted the reproduction of the diagrams in Figs. 11, 12 and 13 from the Final Report † of the Committee on the Efficiency of Internal-Combustion Engines. They also permitted the curve in Fig. 14 to be reproduced. It shows the heat contents for 1 lb. of air, and the associated quantity of water vapour. It therefore enables the observer to ascertain at once the heat contained in any weight of air at different temperatures. The ability to do this rapidly is very useful when a heat balance sheet is being made out for a gas engine run, and when considering the design of vaporisers for gas producers.

EXAMPLES

1. The mixture in a petrol-engine cylinder at atmospheric pressure and volume 1 is found to be at a temperature of 115° C. It is compressed and ignited. At a certain instant the pressure is 15 atmospheres and the volume 0·25. Find the temperature. [B. of E., 1910.]

2. Dry air is pumped into a closed vessel of constant volume until the pressure inside it is 80 lb. per sq. inch by gauge ; the temperature is 90° F. What will be the pressure in the vessel after it has remained for a considerable time in a room where the temperature is 60° F. ?
[Mech. Sc. Tripos, 1906.]

3. Before compression, on a petrol engine diagram $v = 10$, $p = 15$, temperature = 150° C. At a point on the expansion part of the diagram where $v = 4$, $p = 190$, what is the temperature ? Assume that the mixture behaves as a perfect gas. [B. of E., 1911.]

4. At the beginning of the compression part of the diagram of a gas-engine cylinder, the pressure is represented by a distance 0·31 in. and the volume by 3 in. The temperature is known to be 120° C. At a point on the expansion part of the diagram where the pressure is 7 in. and the volume 0·6 in., what is the temperature ?
[B. of E., 1909.]

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* Mr. A. E. Porte in *Proc. I.E.E.*, 1907.
† *I.C.E. Proc.*, Vols. 162 and 163.

5. One lb. of air has a volume of 4 cu. ft. and a pressure of 50 lb. per sq. inch; the temperature is 127° C. It receives 250 C.H.U., its volume remaining constant. What is its new temperature and pressure? The mean specific heat at constant volume may be taken as 0·17. [B. of E., 1909.]

6. A balloon of 5,000 cu. ft. capacity is to be so far filled with hydrogen at a pressure of 30 inches of mercury and 15° C. that, after ascending to a height where the pressure is 20 inches of mercury and the temperature 0° C., the silk envelope is then fully distended, no gas having been spilled. Calculate the mass of hydrogen required and its original volume. The density of hydrogen is 0·0056 lb. per cu. ft. at normal temperature and pressure. [Mech. Sc. Tripos, 1912.]

7. Air at atmospheric pressure and at a temperature of 70° C. is contained in a cylinder of 2 cu. ft. volume, closed by a piston. The latter is forced down until the air is compressed into $\frac{1}{2}$ cu. ft. Find its resulting pressure (lb. per sq. inch) and temperature, if the compression is performed

(*a*) Very slowly;
(*b*) Very quickly (i.e. so that heat has no time to escape) [$\gamma =$ 1·41].

8. Assuming that the compression curve follows the law $PV^\gamma =$ constant [$\gamma = 1{\cdot}4$], calculate the pressure of the charge at the end of compression, given that the pressure at the beginning of the stroke is 12 lb. per sq. inch abs. and that the final volume is $\frac{1}{5}$ the initial volume. [B. of E., 1912.]

9. A quantity of air at temperature 15° C. and pressure 25 lb. per sq. inch abs. is adiabatically compressed to one-half its volume. Find the resulting pressure and temperature. [Mech. Sc. Tripos, 1911.]

10. Air at 68° F. and atmospheric pressure is compressed adiabatically to 4 atmospheres. It is then cooled at constant volume in a receiver down to initial temperature, and then expanded in a non-conducting cylinder to atmospheric pressure. Find the highest and lowest temperatures.

11. The ratio of compression in the cylinder of a Diesel oil engine is 15 : 1, and the temperature of the air at the end of the suction stroke is 70° C. Assume that the actual law of compression is $PV^{1\cdot3} =$ constant, what is the temperature of the air at the end of compression?

12. A vessel is exhausted of air to a pressure of 12 lb. per sq. inch abs., the pressure of the atmosphere being 15 lb. per sq. inch abs. The temperature of the whole being that of the atmosphere (60° F.), a cock is opened and air allowed to rush in until the pressure is equalized. Assuming that no heat is lost to the walls of the vessel, find the rise in temperature of the air within it. [Mech. Sc. Tripos, 1905.]

13. If a quantity of gas expands isothermally from pressure P_0 lb. per sq. ft., volume V_0 cu. ft., to a place where the pressure and volume

become P_1 and V_1 respectively, show that the work done in ft.-lb. is given by

$$2{\cdot}3026\ P_0V_0\log_{10}\frac{V_1}{V_0}$$

14. A quantity of gas expands, the pressure (in lb. per sq. ft.) and volume (cu. ft.) being connected by the law PV^n = constant. The initial pressure and volume being P_0 and V_0 and the final pressure and volume P_1 and V_1 show that the work done by the gas is

$$\frac{P_0V_0-P_1V_1}{n-1}\ \text{ft.-lb.}$$

Show also that the number of heat units received by the gas is

$$\frac{P_0V_0-P_1V_1}{J}\cdot\frac{\gamma-n}{(\gamma-1)(n-1)}$$

and hence show that if the curve PV^n = constant lies below the adiabatic curve passing through the point (P_0, V_0), the gas must be rejecting heat.

15. A pound of air at atmospheric pressure and at 20° C. is to be compressed adiabatically to 10 atmospheres. Find the work done by the pump. The same result is arrived at by isothermal compression, cooling the air so that it keeps at 20° C., and when the pressure reaches 10 atmospheres it is heated at constant pressure. Take the specific heats of air as 0·238 and 0·1694. State separately the work done upon and by the air, and the heat taken from and given to it, all in ft.-lb.

16. A cartridge containing 4 lb. of air at 1,000 lb. per sq. inch (gauge pressure) and 15° C. is placed in the chamber of a gun behind a light frictionless piston fitting the bore of the gun. The cartridge is perforated and the piston just reaches the muzzle of the gun. Calculate the final temperature of the air and the volume of the gun, on the assumption that the air absorbs no heat from the walls of the gun. [Atmospheric pressure = 14·7 lb. per sq. inch.]

[Mech. Sc. Tripos, 1912.]

17. A torpedo air-chamber contains initially 80 lb. of air at a pressure of 1,700 lb. per sq. inch abs. and 15° C., and at the end of the run the pressure is 500 lb. per sq. inch and the temperature 2° C. How much of the heat of the air which is left in the chamber has been abstracted from the sea? [Mech. Sc. Tripos, 1911.]

18. During the inflation of a balloon with hydrogen, the envelope breaks away when only $\frac{4}{5}$ full. It rises in the air so quickly that there is no time for heat to enter or escape through the envelope. What will be the temperature of the hydrogen by the time it has expanded so as to fill completely the balloon, and what will be the barometric height of the altitude at which this occurs?

Temperature of H. on ground = 60° F.
Barometric height on ground = 30 inches of mercury.
Specific heat at constant pressure of H = 3·38.
Specific heat at constant volume of H = 2·38.

19. A cubic foot of air at atmospheric pressure is compressed to 5 atmospheres according to the law $PV^{1\cdot45}$ = constant. Initial temperature = 59° F.

Find—

(i) Work done during compression.

(ii) Heat received or rejected.

(iii) Final temperature and volume.

20. One pound of air is at 2 atmospheres and at a temperature of 20° C. How many cu. ft. does it fill ? It receives heat energy equivalent to 100,000 ft.-lb., its volume remaining constant. Find the new temperature and pressure. The mean specific heat at constant volume of the air may be taken as 0·17.

21. Liquid fuel is burnt in the air supply of a compressed air engine in the proportion of 1 lb. of fuel to 100 lb. of air, and the arrangements are such that the pressure is kept constant. Assuming that the calorific value of the liquid fuel is 20,000 B.Th.U.'s per lb. and that the specific heat at constant pressure of products of combustion is the same as that of air, viz., ·238, what will be the temperature of the heated "air" entering the cylinder, if the temperature of air and fuel before combustion was 60° F. ?

22. Air expands under a piston from a volume of 1 cu. ft. and pressure 300 lb. per sq. inch abs. to volume 5 cu. ft. and pressure 40 lb. per sq. inch abs. Assuming that the pressure and volume vary during the expansion according to the law PV^n =constant, find the heat absorbed in B.Th.U.

23. A gas engine works with an ideal substance of constant specific heat, receiving and rejecting heat at constant volume and with adiabatic compression and expansion. The piston displacement per stroke is 1·2 cu. ft. and the clearance volume 0·15 cu. ft. Calculate the theoretic thermal efficiency of the engine, taking γ as 1·38.

24. The cycle of operations in a gas-engine is as follows :—

(1) Gas is compressed from V=3·76 cu. ft. to V=0·6 cu. ft. according to the law $PV^{1\cdot4}$= 94·5. [P being the pressure in lb. per sq. inch.]

(ii) On explosion, the pressure rises to 420 lb. per sq. inch, the volume remaining constant.

(iii) Expansion takes place till V = 3·76 once more, the law followed being $PV^{1\cdot2}$ = constant.

(iv) The pressure falls to its initial value, the volume remaining 3·76.

Draw out the PV diagram from these data and find the mean effective pressure in lb. per sq. inch.

25. A gas-engine works on an ideal cycle, with adiabatic compression and expansion, receiving and rejecting heat at constant volume. The piston displacement per stroke is 1 cu. ft., the clearance volume 0·2 cu. ft., and at the beginning of compression the temperature of the cylinder contents is 600° F. absolute, the pressure being atmospheric. The engine receives 0·06 cu. ft. of gas per cycle (calorific value 600 B.Th.U. per cu. ft.). Atmospheric pressure = 14·7 lb. per sq. inch.

Find—

(i) Weight of cylinder contents.

(ii) Pressure and temperature at end of compression [$\gamma = 1{\cdot}38$].

(iii) Rise of temperature during explosion [neglect jacket loss and take $C_v = 0{\cdot}18$].

(iv) Pressure at end of explosion.

(v) Temperature and pressure at end of expansion.

(vi) Efficiency of the cycle.

(vii) Efficiency of an engine working on a Carnot cycle between the same highest and lowest temperatures.

[Mech. Sc. Tripos, 1906.]

26. In a gas engine trial, the curve of expansion was $PV^{1{\cdot}2} = k$ and of compression $PV^{1{\cdot}35} = k$. The remainder of the heat was received and rejected at constant volume. The highest and lowest temperatures were 2250° F. and 520° F. The piston displacement was 2·8 cu. ft. and the clearance 1 cu. ft.; $\gamma = 1{\cdot}35$. The initial pressure was atmospheric.

Find—

(i) Temperatures at end of compression and expansion.

(ii) Heat received and rejected during each operation.

(iii) Efficiency of engine.

27. An air-pump working in an airship takes air direct from the atmosphere where the pressure is 10 lb. per sq. inch. The inlet valve closes at the completion of the suction stroke, and the pressure is then just equal to that of the atmosphere. The mean pressure in the pump during the suction stroke is 2 lb. per sq. inch below that of the atmosphere. Neglecting clearance, and assuming that no heat passes between the cylinder walls and the cylinder contents during the suction stroke, show that the volume of air (reckoned at external temperature and pressure) taken per stroke is 5·7 per cent. less than the stroke volume. The volumetric heat of air may be taken as 19·5 ft.-lb. per standard cu. ft. [Mech. Sc. Tripos, 1913.]

28. A single-stage compressor is used to maintain the pressure in a receiver at 1,500 lb. per sq. inch while air is being drawn from the receiver. Compare the work done per lb. of air if the law of compression is $pv^{1{\cdot}3} = k$ with that done if the compression is isothermal. Specific volume of air at atmospheric temperature and pressure = 13·1 cu. ft. per lb.

29. A building is to be heated by passing the air for ventilating it through a compressor which compresses it adiabatically, then throttling it down to atmospheric pressure on leaving the compressor. The air enters the compressor at 0° C. and leaves it at 25° C., and the flow of air is 80 lb. per minute. Calculate the power required (assuming unit mechanical efficiency).

If the power thus usefully employed costs ½*d*. per k.w. - hour, compare the cost of heating thus with that when burning coal in a stove, the iron chimney of which passes through the air-duct and imparts 20

per cent. of the heat of combustion to the ventilating air. The cost of coal is $0 \cdot 1d.$ per lb. of calorific value 8,000 C.H.U. per lb.

[Mech. Sc. Tripos, 1911.]

30. Air is compressed adiabatically into a receiver of V cu. ft. capacity to m times the atmospheric density. Show that, if P be the atmospheric pressure in lb. per sq. ft., the work expended is

$$PV\left[\frac{m^\gamma - m}{\gamma - 1} + 1\right] \text{ ft. lb.}$$

[Mech. Sc. Tripos, 1904.]

31. Show that when a perfect gas is wire-drawn from one pressure to a lower one, without any gain of kinetic energy, the temperature is unaltered after expansion.

32. In an ideal diagram of a Diesel engine, the gas is compressed adiabatically from volume V_1 to V_2, then expands from volume V_2 to V_3 at constant pressure, further expands adiabatically from V_3 to V_1 and finally rejects heat at constant volume V_1. Show that the thermal efficiency may be expressed as

$$1 - \left(\frac{1}{r}\right)^{\gamma - 1} \cdot \frac{R^\gamma - 1}{\gamma(R - 1)}$$

$$\text{where } r = \frac{V_2}{V_1} \text{ and } R = \frac{V_3}{V_2}.$$

CHAPTER III

Combustion and Explosion

Chemical Combustion—Dugald Clerk's and Grover's Early Experiments on Explosion in Closed Vessels—Discussion of Results—Increase of Specific Heats of Gases—Dissociation—"After-burning"—Later Explosion Experiments—Time of Explosion—Turbulence—Gaseous Explosions Committee.

44. Chemical Combustion.—Instances of chemical combustion are manifold. Two among the commonest are the burning of coal, and the oxidation of the carbon in food which is the source of the heat energy given out by the human body. In place of coal, it is possible to burn gas made from coal and so obtain either heat or light. In a gas engine cylinder, gas and air are first mixed together and the whole mass ignited at once, so that the union is explosive. Useful figures to remember are that 1 lb. of coal on being burnt will liberate about 12,000,000 ft.-lb. of energy, a cubic foot of coal gas will liberate about 550,000 ft.-lb., 1 lb. of petroleum about 18,000,000 ft.-lb., and 1 lb. of petrol some 15,000,000 ft.-lb. These are very large amounts, and were it possible to invent a heat engine of 100 per cent. efficiency it is plain that a very liberal supply of energy would be obtainable at little cost. With existing engines 1 lb. of coal with potential energy equal to 12,000,000 ft.-lb. will only give in energy on the brake about 3,000,000 *ft.-lb. with the best steam engines* and 4,000,000 *ft.-lb. with the best gas engines,* the waste energy being 9,000,000 ft.-lb. and 8,000,000 ft.-lb. respectively in the two cases.

The loss of 8,000,000 ft.-lb. which occurs in a gas engine is divided between the loss to the water in the cooling jacket and the loss which occurs owing to the exhaust products

being at a high temperature and so carrying off a large unutilized portion of the heat. The loss to the cylinder walls is the more difficult to follow in all the intricacies of the working cycle. The cooling jacket is necessary,* as without it the piston and cylinder would get almost red hot and the engine would stop running. The temperature-flow through the metal depends on the position of the piston in its stroke, but it is difficult to determine the precise relationship.

45. If, after a charge of gas and air has been drawn into a gas engine cylinder, the flywheel be held so that it cannot move and the charge be then ignited, a rapid rise of pressure is recorded on the indicator. It ought, one might think, to be easy to calculate what this rise would be, since the quantity of gas and air admitted and their quality are easily determinable and the amount of thermal energy liberated is therefore known. If this amount of energy be divided by the amount of heat required to heat the mixture through one degree Cent. it is clear that the resulting temperature would be ascertained, and from this it would be simple by the $\frac{PV}{T}$ law to determine the resulting pressure. This had often been done, but it had always been found that the pressure actually obtained was only **about one-half** that calculated. Here are the actual figures obtained in some experiments carried out many years ago by Dugald Clerk—

Ratio air/gas	Absolute Pressure Obtained	Absolute Pressure Calculated
	lb. per sq. inch	lb. per sq. inch
14	55	110
13	$66\frac{1}{2}$	116
12	75	123
11	76	132
9	93	161
7	102	190
6	105	214
5	106	206
4	95	196

* Except in very small cylinders, which are sometimes air cooled.

On an average there appears here to be a loss of as much as 50 per cent. of the pressure. Why is this ?

46. Several **explanations** have been put forward to account for this loss. The most important are—

1. *The Dissociation Theory.*—It is well known that chemical compounds such as H_2O or CO_2 dissociate at high temperatures into simpler gases and in so doing absorb heat. It has therefore been thought that at the high temperatures of explosion such dissociation would occur and the heat so absorbed might account for the missing 50 per cent. This assumption, however, involves the deduction that for weak explosions, in which low pressures and temperatures were attained, the effect should be much less, so that the actual pressure would form a much larger proportion of the calculated pressure, and the converse in the case of rich mixtures. As a glance at the above figures will show, this, however, is not the case; at the weakest mixture of 1 to 14 the missing pressure is 50 per cent., and at the richest of 4 to 1 it is 52 per cent., or practically the same. This theory alone therefore does not suffice to account for the observed facts.

2. *The Cooling Theory.*—This assumes that the cooling effect of the cylinder walls is so great that the pressure actually obtained must fall much below the ideal calculated. It does not explain, however, why the loss should be always 50 per cent. in the particular cylinder used, nor, moreover, does it explain why a 50 per cent. loss is found still to occur even when a cylinder of a different size and shape is chosen. So that this theory also is inadequate in itself to explain the observed effect.

3. *The Increasing Specific Heat Theory.*—This is the theory advanced first by MM. Mallard and Le Chatelier, who found as the result of their experiments that the specific heat of gases and particularly of CO_2 appeared to increase considerably with rise of temperature. The objection commonly alleged against this theory is that, as in the *Dissociation Theory*, it requires that a greater proportion of the ideal pressure should be obtained at lower temperatures than at higher, and that this is not found to be the case.

4. *The After-burning Theory.*—This theory has chiefly

been associated with the name of Dugald Clerk, who suggested that the combustion of the gas was not as rapid as supposed and that not all the heat was liberated before the moment of highest pressure. It assumed in fact that the gas was still burning long after the point of maximum pressure and that the cooling effect of the walls had therefore a much longer time to operate than had been generally supposed. In an actual gas engine this would mean that the gas would be burning right through the working stroke, and that it must sometimes happen that unburnt gas would pass away in the exhaust. The objection to this theory lies in the fact that it has never been shown conclusively that the explosion is not complete at the point of highest temperature. Indeed the evidence is rather the other way. It is not usual to find that the exhaust contains more than a very few per cent. of unburnt gases, and it has moreover been shown that a complete heat balance analysis can be obtained without the need of any such hypothesis.

47. Thus there are four simple theories, of which none appear to be sufficient in themselves to account for the observed loss. The difficulty is so fundamental a one that still further theories compounded of the above have been put forward. Dugald Clerk made the suggestion, as will be explained later at greater length, that the "suppressed" 50 per cent. may be accounted for on the supposition that part is due to the after-burning loss and part to a certain increase in specific heats. The author has seen no reason to modify the suggestion he put forward at the meeting of the British Association * in 1902, viz. that the so-called "suppression of temperature" is probably due to the combined action of cooling and of increase of specific heat on the lines suggested by the French physicists, MM. Mallard and Le Chatelier. Although the increase of specific heat left a larger proportion of loss to be accounted for at low temperatures than at high ones, this was sufficiently explained by the fact that the ignition period was much longer at low temperatures and so allowed the cooling effect to have a longer time for action than it would have

* *The Engineer*, October 10, 1902; *Engineering*, October 10, 1902.

at high temperatures. This meant that for weak mixtures the 50 per cent. loss was mainly due to cooling, for rich mixtures mainly due to increase of specific heat, and for intermediate mixtures was due to a combination of the two.

Dugald Clerk's early experiments consisted in indicating explosions of mixtures of air with Glasgow and Oldham gas in a closed cylinder 7 in. by $8\frac{1}{4}$ in. The indicator registered pressure p on a rotating drum driven at a known constant speed, so that curves were obtained showing the relation between p (pressure) and t (time) during the explosion and the subsequent cooling of the gas to the walls and ends of the cylinder. From the diagrams so obtained it was of course possible to measure the time occupied by the explosion, and the subsequent rate of fall of pressure due to cooling. At the time these experiments were made the specific heat was thought to be constant and it is important to note how greatly its now known increase with temperature affects the calculated pressures, particularly if for the moment MM. Mallard and Le Chatelier's figures be adopted. That there are objections to the method of experimenting by which the French physicists obtained their results is well known. In fact Prof. Callendar has remarked: "The method of experiment employed was closely analogous to the explosion that was taking place in the gas engine itself. Explosive mixtures were fixed in a closed cylinder 17 in. by 7 in., and the maximum pressure was read by means of a Bourdon gauge." Since the date of their experiments other measurements have been made, and these will in due course be discussed; but the increase of specific heat with temperature is undoubted, and for the present purpose MM. Mallard and Le Chatelier's figures are taken as illustrative. If the theoretical temperature of explosion is calculated from these values the difference from the observed value is much less. Thus a column may now be added to the table last given and the results are also shown in Fig. 15.

It will be seen that in the case of the weakest mixture the 50 per cent. loss has been reduced to 34 per cent., and in the case of the richest 52 per cent. has been reduced to 23 per cent., showing a step in the required direction. The balance is

Ratio air/gas	Absolute Pressure Obtained	Absolute Pressure Calculated on Constant Specific Heat	Absolute Pressure Calculated on Variable Specific Heat
14	55	110	83
13	66½	116	86
12	75	123	90½
11	76	132	95
9	93	161	107
7	102	190	121
6	105	214	131
5	106	206	127
4	95	196	123

taken to be made up of convection and radiation losses. The convection loss to the cold walls of the containing vessel

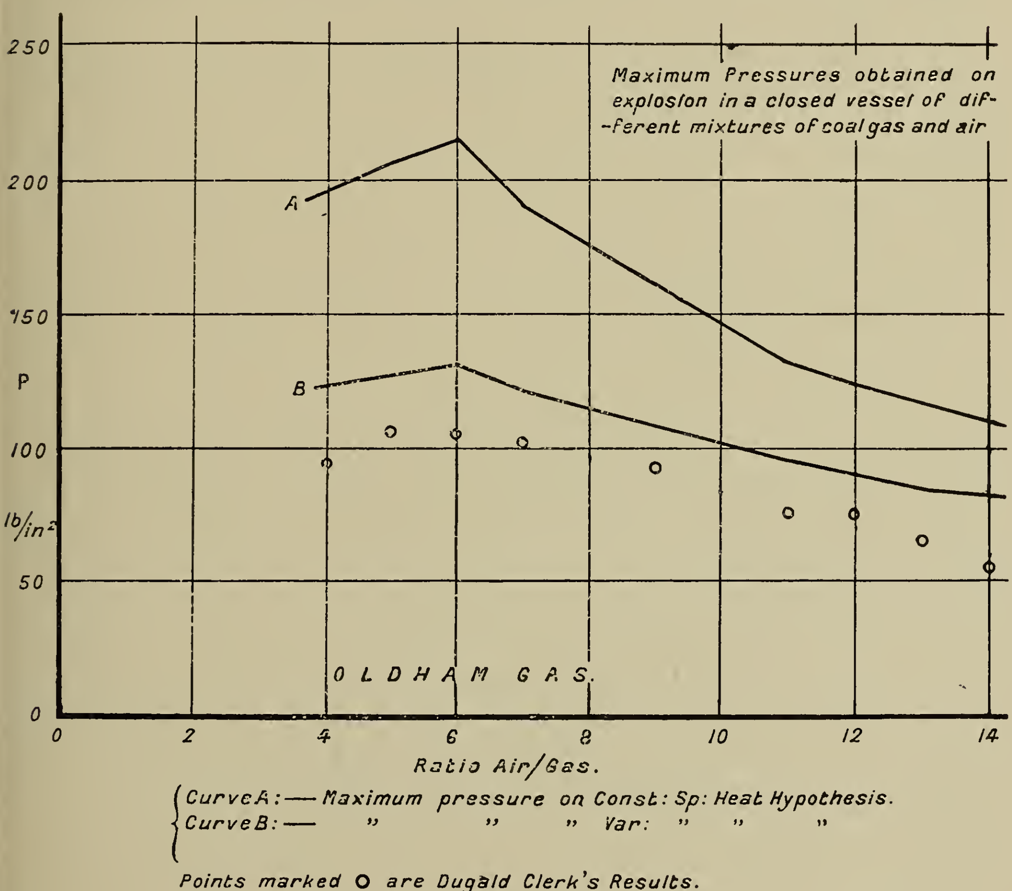


FIG. 15.

takes place at all temperatures; the radiation loss on the other hand is important only at the moments of highest temperature. Prof. Hopkinson has shown that the cooling losses depend also on the state of the inner surface of the vessel, whether bright or blackened.

The law of cooling of gaseous mixtures enclosed in metal cylinders of stated dimensions is not easy to apply to the case of ordinary gas engines. First, because the connexion between the rate of loss of heat and the dimensions of the cylinder is very complicated, but even more because in an ordinary gas engine cylinder the temperature of the cylinder walls and of the piston are so very different that conditions sometimes arise in which while the gas is being heated by the piston it is at the same time being cooled by the cylinder walls, a condition of affairs in no way analogous to that holding in the above experiments.

48. Grover's Explosion Experiments.—At the time these calculations were made the only other well-known experiments upon the explosion of gases in closed vessels were those of **Grover**, and at the British Association meeting in 1903 an endeavour was made to show how far the combined variable specific heat and cooling theory would go towards explaining the very remarkable results obtained by Grover, which in no way resembled those obtained by Dugald Clerk, inasmuch as the former found much lower pressures and came to the unexpected conclusion that the retention of waste products in a gas engine cylinder increased the pressure of the ensuing explosion, an astonishing result having regard to the great care taken by most gas engine manufacturers to sweep out the greatest possible amount of the products of old explosions. The great difference between the maximum pressures obtained by Dugald Clerk and Grover is illustrated in Fig. 16.

It was Grover's idea not only to measure the pressures produced by various richnesses of mixture of coal-gas and air, but to investigate whether the resultant pressure on explosion was affected by replacing the air in excess of that calculated as chemically necessary for complete combustion by a portion of the burnt products of the previous explosion. Now it appears from Grover's account of the

experiments that he had an iron cylinder of one cubic foot capacity, and that in each series of experiments the volume of the coal gas admitted was kept constant and the cylinder was then filled with a mixture of air and waste products in various proportions. This was done in each series by filling the cylinder with water, and allowing gas to enter whilst a known volume of water was run out. Thus after an explosion water was allowed to pass into the cylinder until all but the required volume of burnt products had been forced out; so that if it were desired that no burnt products should be left, the cylinder would be completely filled with water, but if, say, 50 per cent. of the volume of the cylinder was required to contain burnt products, the water would only be permitted to rise half-way up the cylinder.

The pressure was recorded in the customary manner on a rotating drum, but very few of the curves are given in the published account of the experiments, and it is therefore difficult to make a very exact comparison between the time rate of fall of the pressure after explosion in Grover's experiments (using, of course, those experiments in which no burnt products were admitted) with those of Dugald Clerk. However, so far as the curves can be examined, they show for the same pressures almost exactly the same rate of fall, a result which is the less unexpected, as the diameters of the two cylinders appear to have been nearly equal. It is not difficult to calculate what the ideal maximum temperature and corresponding pressure of explosion would be, using the same variable specific heat figures, and assuming no cooling of the gas by the walls, and when this has been done, it may be compared with the pressure found experimentally. The following table (see page 52) shows the result of such a calculation.

In this table there is also given the difference in the heat energy between the gas at this temperature and at the actual temperature attained.

The curves (p. 52) show the actual pressures plotted with respect to richness of mixture for the experiments of both investigators. It is seen that Grover's curve lies far below Dugald Clerk's. This cannot be due entirely to the different

Ratio of Air to Gas	Pressure Observed	Pressure Calculated	Corresponding Difference in Energy (Approx.)
	(Absol.)	(Absol.)	Ft.-lb.
15	31	73	21.000
14	39	76	19.000
13	46	79	18,000
12	51	83	18,000
10	63	92	18,000
8	77	104	18.000
6	77	119	31.000

cylinder volumes used (317 and 1.728 cubic inches), or to differences in the chemical constitution of the gases, because, as will be seen from the intermediate curve, there is little

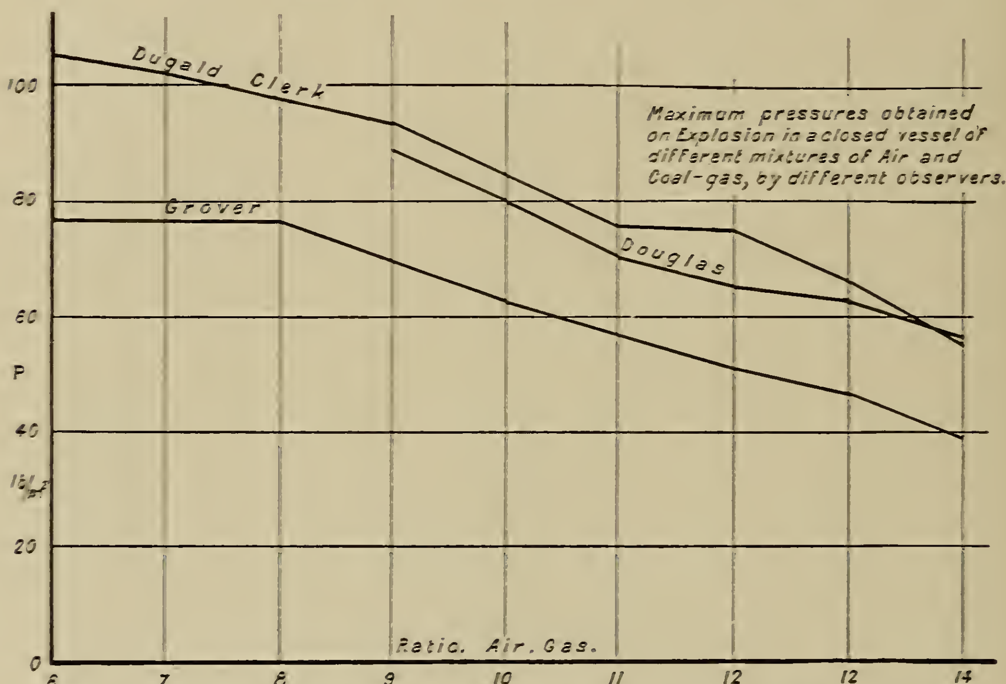


FIG. 16.—Explosion curves showing much lower pressures obtained by Mr. Grover than by other observers.

disagreement between the results obtained by Dugald Clerk and by Douglas, although the results * obtained by the latter were for gases enclosed, not in iron cylinders, but in a eudio-

* See *The Engineer*, April 22, 1887, and November 7, 1902.

meter tube. If the use of a eudiometer does not produce results more different from Dugald Clerk's than this, the presumption certainly is that some factor must have entered into Grover's experiments which has entirely masked his results. A suggestion as to what this factor could be has been made by Grover himself, for in describing one experiment he says: "The difference is no doubt due to the fact that water was present on the walls of the cylinder"; but Grover did not consider apparently that this presence of water affected his conclusions on the subject generally—conclusions which are set forth in his *Modern Gas and Oil Engines*.

There is, of course, a limit to the quantity of water which could adhere to the walls of the cylinder, and it is necessary to see whether the required amount is what could reasonably be expected to exist. The average loss of energy given in column 4 of the table on p. 52 is about 20,000 ft.-lb, and considering the average energy given to 1 lb. of water to raise it from atmospheric temperature to superheated steam at the average maximum temperature, as about 750 Cent. heat units, it follows that the weight of water required equals 0·0191 lb. This would occupy a space of about 0·53 cubic inch, and in a cylinder of the dimensions used a film of water $\frac{1}{2000}$ in. thick would be sufficient to account for this. So that there is no difficulty in accounting for the presence of a sufficient quantity of water. To show the result of the presence of water a simple example may be taken, in which the weight of the water film is three-eighths of the weight of the gaseous mixture, and in which both continue at the same temperature and pressure. When the pressure (absolute) amounted to 33 lb. per square inch, a calculation made in the absence of the knowledge of the presence of a water film would give a temperature of 254° Cent., whereas the real temperature would be 124° Cent., a very different result. A further calculation with the same amount of water present shows that a pressure of 60 lb. per square inch would be attained on explosion, whereas under the same circumstances, but in the absence of the water film, a pressure of 100 lb. per square inch would have been attained.

It may be concluded that the presence of a water film of

varying extent is a sufficient explanation of the very curious results obtained by Grover.

49. Later Experiments.—In addition to the early experiments of Dugald Clerk and Grover, some work in the same direction was done at the Massachusetts Institute of Technology, but it does not appear that any definite conclusions were drawn therefrom. Later experiments have been made by Dugald Clerk, Hopkinson, and Bairstow and Alexander at the Royal College of Science. Taking the last first: Bairstow and Alexander's experiments were made on mixtures of London coal gas and air in a cylinder 18 in. long and 10 in. in diameter, pressures were indicated on a rotating drum, and the results of the investigations were communicated to the Southport meeting of the British Association in 1903, and to the Royal Society two years later. The chief interest of these tests lies in the fact that various initial pressures were used, instead of the atmospheric initial pressure used by earlier experimenters. The following table gives a selection from their results—

ROYAL COLLEGE OF SCIENCE EXPERIMENTS. EXPLOSIONS IN CLOSED VESSEL; VARIOUS INITIAL PRESSURES AND TWO MIXTURE STRENGTHS.

Mixture		Initial Pressure lb. per sq. in. abs.	Initial Temp. °C.	Max. Press. observed lb. per sq. in. abs.	Time to reach this pressure. secs.
Air	Gas				
1	0·169	44·8	23·5	348	0·042
1	0·172	34·5	22	270	0·041
1	0·170	24·7	21	189	0·041
1	0·168	14·55	21	112	0·036
1	0·166	9·71	24·5	68	0·05
1	0·166	7·18	24	47	0·10
1	0·103	44·7	16·5	238	0·33
1	0·105	34·6	18·6	185	0·35
1	0·104	24·7	20·0	126	0·41
1	0·107	14·4	21·0	74	0·44
1	0·104	9·5	21·5	46	0·50
1	0·107	7·06	22·0	33	0·50

Details of these experiments will be found in Dugald Clerk's book on *The Gas Engine*, Vol. I. It is left as an exercise to students to compare these explosion pressures with those theoretically obtainable, taking the Gaseous Explosions Committee's figures for specific heats.

50. Time of Explosion.—In 1900 Dugald Clerk measured the time occupied in the explosion of coal gas and air from the moment of ignition to that of maximum pressure. The following were some of the times for various ratios of air to gas.

Volumes of air to one volume of gas	Time of explosion
11	0·290 sec.
9	0·155 ,,
7	0·067 ,,
6	0·055 ,,

These figures were obtained from explosion experiments in closed vessels in which the mixture before explosion was at

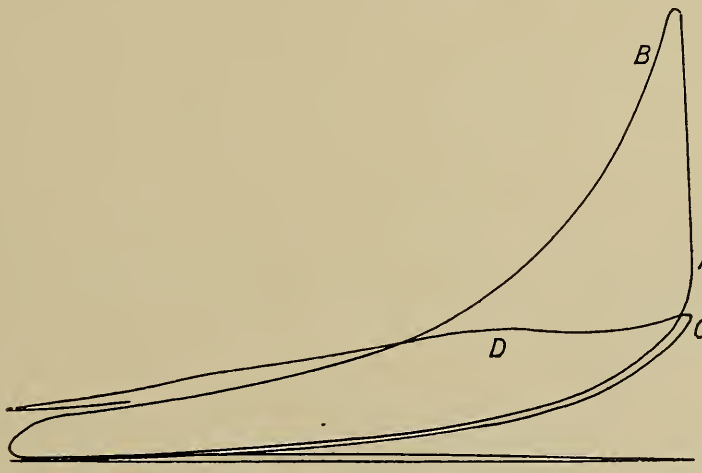


FIG. 17.—Indicator diagrams showing effect of Turbulence. A B is explosion curve when charge is fired on first compression stroke; duration of explosion 0·037 second. C D is explosion when charge is fired upon the third compression; duration of explosion 0·092 second.

rest. They may be compared with those in the table on p. 54. When the mixture is turbulent the time of explosion is

very much shorter. In a working gas engine for instance the times of explosion are much less than any of the above figures. Indeed did the explosion take as long as in the above table, the working of the engine would be quite impossible. A small gas engine can be made to run at 600 r.p.m. so that each stroke occupies only 0·05 sec., and the explosion is usually completed in quite a small fraction of the stroke, which shows how much more rapid the explosion must be when there is turbulence.

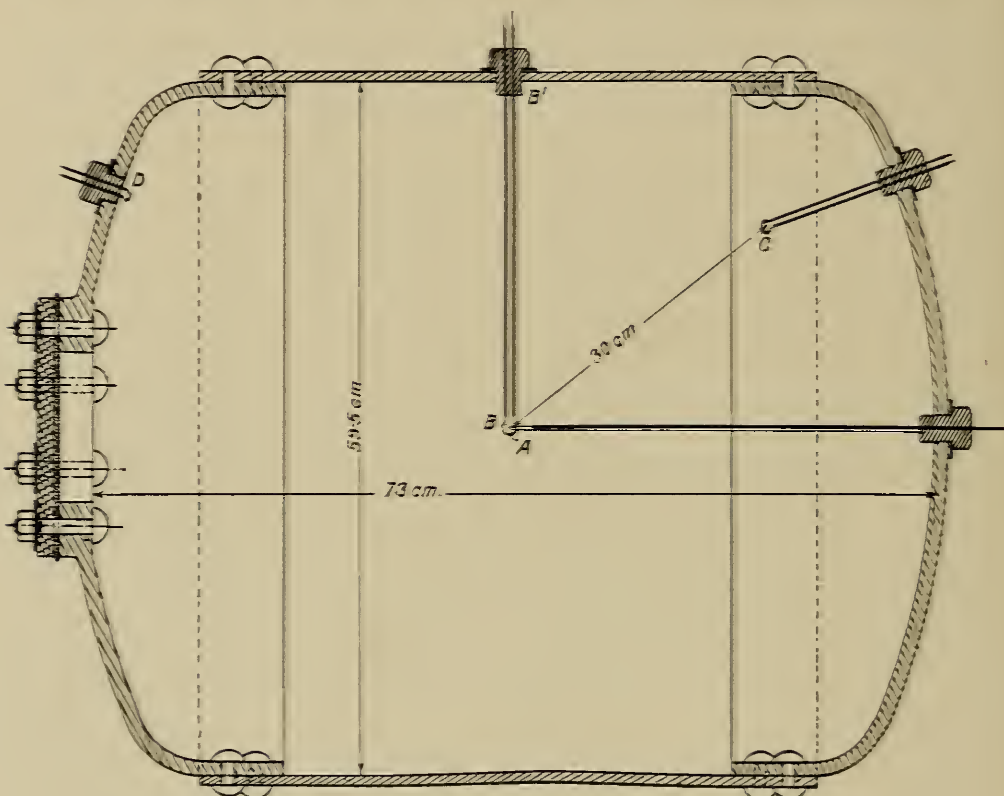


FIG. 18.—Professor Hopkinson's Gas Explosion Apparatus.

The effect of turbulence is well shown in Fig. 17, which is reproduced from some indicator cards taken by Dugald Clerk. In the one case we have the normal diagram produced by normal ignition arrangements, whilst in the other the turbulence which the gaseous mixture had during the suction stroke has intentionally been given time to die down before the spark is passed. The effect of this delay is seen in the changed diagram, which illustrates clearly the continuance

of combustion throughout the whole length of the expansion stroke.

51. Hopkinson.—A series of explosion experiments was undertaken by Prof. Hopkinson and communicated to the Royal Society in 1906. The vessel is shown in Fig. 18. A is the sparking point, B, C and D are platinum thermometers. Thermometer B is practically at the centre of the vessel, C is about 30 cm. distant from the spark and D is about 1 cm. from the walls of the vessel. A record of the pressure was taken on the same drum as that upon which the temperatures were electrically recorded. The indicator* was very simple, consisting as it did of a piston controlled by a flat steel spring held at the two ends. As the spring was deflected a mirror tilted and so threw a beam of light on to the moving film. The period of the instrument was about $\frac{1}{300}$ sec. Fig. 19 shows the result obtained in the form of a graph. The following table serves also to show the actual indications recorded by the electric thermometer placed at the centre of the vessel :—

Time Secs.	Resistance Ohms	Rise of Resistance Ohms	Temperature in Degrees C.
0·008	22·05	12·4	560
0·024	30·3	20·7	995
0·041	32·7	23·1	1,135
0·057	33·1	23·5	1,165
0·074	33·1	23·5	1,165
0·09	34·0	24·4	1,225
0·107	34·5	24·9	1,260
0·123	34·7	25·1	1,275
0·140	34·7	25·1	1,275
0·173	36·6	27·0	1,400
0·26	wire melts	—	1,710

An investigation had also to be made into the question of the existence of a time and temperature lag in the temperature recorded by the thin platinum wire. Prof. Hopkinson found the temperature of the wire to lag materially behind that of the gas when the latter was changing rapidly. To measure this, wires of two different thicknesses were used, viz. $\frac{1}{1000}$

* *See* Fig. 40.

in. and $\frac{1}{2000}$ in. respectively, and by a comparison of the results obtained Prof. Hopkinson was able to find the amount of the correction which he considered it necessary to employ.

The most important of the conclusions reached by this experimenter, who, he tells his readers, carried out these experiments, largely "with the object of finding the cause of the so-called 'suppression of heat' in explosions," is that his experiments appear to prove that *even in the weakest mixtures, combustion, when once initiated at any point, is almost*

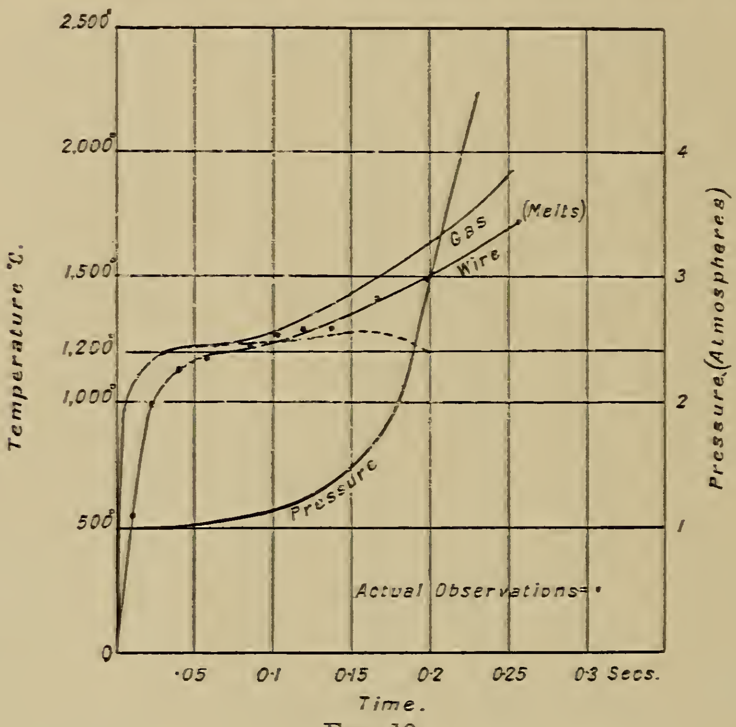


FIG. 19.

instantaneously complete. Moreover, he adds, they show that the specific heat of the products is very much greater at high temperatures than at low, and the extent of the difference seems to justify the view that it is the main reason of the so-called "suppression of heat."

52. In addition to these conclusions Prof. Hopkinson found certain *differences in the temperature of the gas in different parts of the vessel*, and this supports the results obtained by Prof. Burstall in his gas engine trials for the Institution of Mechanical Engineers. In experimenting with a rich mixture

(air/gas = 9) Professor Hopkinson found that at the moment of maximum pressure the distribution of temperature in his vessel was roughly as follows—

Mean temperature (inferred from pressure) .	1,600° C.
(*a*) Centre near spark	1,900° C.
(*b*) 10 cm. within the wall (C, Fig. 11) . .	1,700° C.
(*c*) 1 cm. from wall at end (D, Fig. 11)	1,000 to 1,300° C.
(*d*) 1 cm. from wall at side	850° C.

It is explained that "at points *a*, *b* and *c* the gases can have lost but little heat at this time, and the differences of temperature are almost wholly due to the different treatment of the gas at different places. At (*a*) it has been burnt nearly at atmospheric pressure, and compressed after burning to about 6½ atmospheres absolute, while at (*c*) it has been first compressed to about six atmospheres as in a gas engine, and then ignited without any subsequent compression. At the point (*d*) much heat has been lost, since this is the first point on the wall reached by the flame; the gas here is ignited when the pressure is about two atmospheres, its temperature rises instantly to 1,300° C. and at once begins to fall."

In experiments on a weak mixture of twelve volumes of air to one of gas the explosion was affected very greatly by the convection current set up, owing to the ignited gas being lighter and rising through the vessel. In the rich mixture this could not happen to the same extent, as the maximum pressure was reached about a quarter of a second after firing, whilst with the weak mixture the interval was two and a half seconds and so the time for convection was much longer. The experimenter recorded that "a few centimetres below the spark the temperature will rise rapidly and then fall; the flame reaches the wire, and is then carried upward and away from it, the wire being cooled by the current of cold, unburnt gas which follows in the wake of the ascending flame. About one second after ignition, and while the pressure is still less than 10 lb. above atmosphere, the upper part of the vessel is filled with burnt gas which is in contact with, and losing heat to, the upper half of the walls." The lower half of the gas is therefore burnt last. Finally it may be recorded that Professor Hopkinson in comparing the behaviour

of rich and poor mixtures says: "It is safe to assume in dealing with a 12/1 mixture that one-fifth of a second after maximum pressure (when the loss of pressure by cooling is still less than 5 per cent.) there is present in the cylinder a mass of CO_2, H_2O, and inert gas in complete chemical equilibrium. In the 9/1 mixture this state is, of course, attained very much sooner. The difference in the behaviour of the weak and strong mixtures is wholly due to the very slow propagation of flame in the former; in a 9/1 mixture the flame seems to travel about ten times as fast as in the 12/1 mixture."

Professor Hopkinson has also measured the temperature of the air in a cylinder when the engine is turned by an electric motor. The air is then compressed and expanded almost adiabatically. It was found that at the top of compression the temperature of the air half a centimetre from the wall was some 30 deg. Cent. less than it was in the centre. At points nearer the wall, that is, within 1 mm. the temperature fell off very rapidly—although still materially above that of the wall face even at a distance from it of only $\frac{1}{10}$ mm.

53. The "Zig-Zag" Experiments.—In 1906 Dugald Clerk communicated to the Royal Society the results of some explosion experiments he had made by a new method to determine the volumetric heat of the gaseous mixture used in the gas engine. According to that description this consisted in running a gas engine under the ordinary standard conditions, and then at a given moment preventing the exhaust and inlet valves from opening, and at the same time taking a series of indicator diagrams. These diagrams showed a number of expansion and compression curves with the pressures gradually falling as the gas cooled. Fig. 20 is a representation of a series of curves so obtained. From the shape of such curves it is possible to calculate what is occurring to the gas in the cylinder. The following explanation of the method is given in the First Report of the B. A. Gaseous Explosions Committee: "The calculation is based on the assumption that the total heat loss from the hot gases during any given portion of a stroke is the same in expansion and compression if the mean temperature be the same. In the first compression the temperature of the gas rose to about 1100° C. (at the

point C, Fig. 20). During the first three-tenths of the following expansion stroke (CD), the temperature fell to about 700° C. The work done in this part of the expansion was measured and the heat loss determined as above was added. Thus the change of internal energy corresponding to the temperature change 1100°—700° is obtained. The average volumetric

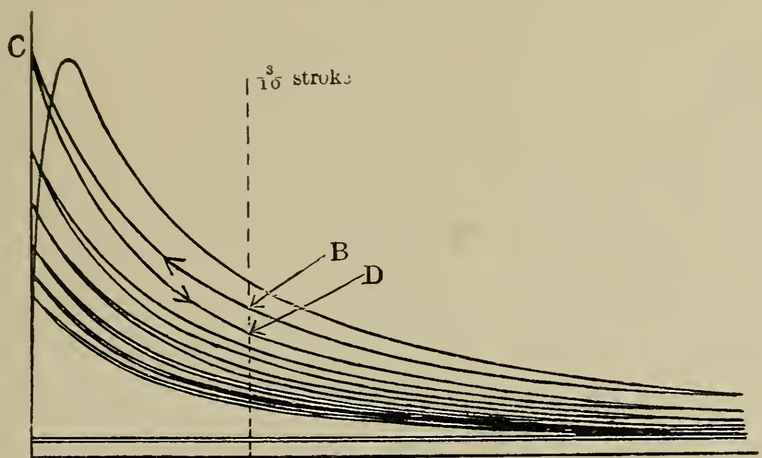


FIG. 20.—Dugald Clerk's "Zig-Zag" curves. 1/9 mixture.

heat over this range is within the errors of experiment equal to the volumetric heat at the mean temperature of 900° C., which accordingly is by this method determined direct instead of by difference, as is necessarily the case when (as in some other experiments) the whole internal energy change associated with complete cooling of the gas is measured."

The volumetric heat figures thus found by Clerk are given in the following tables.

VOLUMETRIC HEAT (INSTANTANEOUS) OF WORKING FLUID.

Temperature	Volumetric Heat	Temperature	Volumetric Heat
Degrees C.	Ft.-lb.	Degrees C.	Ft.-lb.
0	19·6	800	26·2
100	20·9	900	26·6
200	22·0	1,000	26·8
300	23·0	1,100	27·0
400	23·9	1,200	27·2
500	24·8	1,300	27·3
600	25·2	1,400	27·35
700	25·7	1,500	27·45

Mean Volumetric Heat of Working Fluid over Temperature Range shown

Temperature	Volumetric Heat	Temperature	Volumetric Heat
Degrees C.	Ft.-lb.	Degrees C.	Ft.-lb.
0—100	20·3	0—900	23·9
0—200	20·9	0—1,000	24·1
0—300	21·4	0—1,100	24·4
0—400	21·9	0—1,200	24·6
0—500	22·4	0—1,300	24·8
0—600	22·8	0—1.400	25·0
0—700	23·2	0—1,500	25·2
0—800	23·6	—	—

It will be observed from the above that although the apparent specific heat was found to increase with rise of temperature, it tended towards a limiting value. The increase found for the first 500° C. was far more than for the last 500°. This conclusion does not quite accord with the experiments of other workers.

54. Gaseous Explosions Committee.—This Committee was appointed by the British Association in 1907 "for the Investigation of Gaseous Explosions, with special reference to Temperature." No other work has thrown so much light upon the theory of the internal combustion engine as have the labours of this Committee. One of the first tasks undertaken was a thorough sifting of the experimental work bearing on the rise of specific heat of gases with temperature. This experimental work was divided into three classes :—

(1) Constant-pressure experiments : Regnault, Wiedemann, Witkowski, Lussana, Holborn and Austin, Holborn and Henning. The gas is heated from an external source in these experiments, and is at atmospheric pressure.

(2) Experiments in which both volume and pressure are varied, the gas being heated by compression. The above mentioned experiments of Clerk and the determinations of the velocity of sound in hot gas by Dixon and others belong to this class.

(3) Constant-volume experiment. To this category belong

the explosion experiments of Mallard and Le Chatelier, Clerk, Langen, Petavel, Hopkinson, and others, and Joly's determinations with the steam calorimeter. In the explosion experiments the gas is heated by internal combustion.

As a result of a full examination of this large mass of experimental work the Committee published a curve of internal energy at various temperatures for the gas engine mixture with which Clerk had experimented (Air/Gas = 9/1), and this

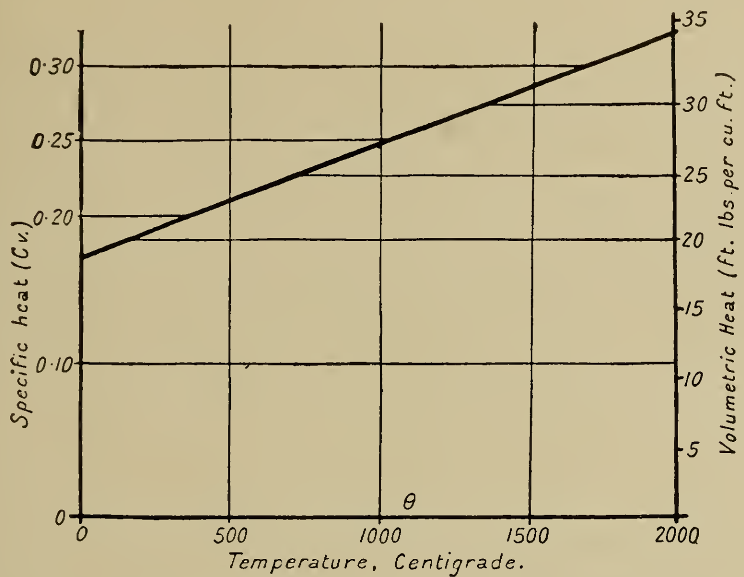


FIG. 21.—Specific Heat, and Volumetric Heat, of expanding Gases in a Gas Engine at different Temperatures.

curve in slightly modified form is reproduced in Fig. 25, on p. 84.

The tangent at any point of this curve is a measure of the specific heat at that point, and it is found that the following linear equation represents the specific heat within the limits of experimental accuracy

$$C_v = 0{\cdot}152 + 0{\cdot}000075\ T$$

or in other units

Volumetric heat $= 16{\cdot}6 + 0{\cdot}0082\ T$ in ft.-lb. per cubic foot.

This is illustrated in Fig. 21,

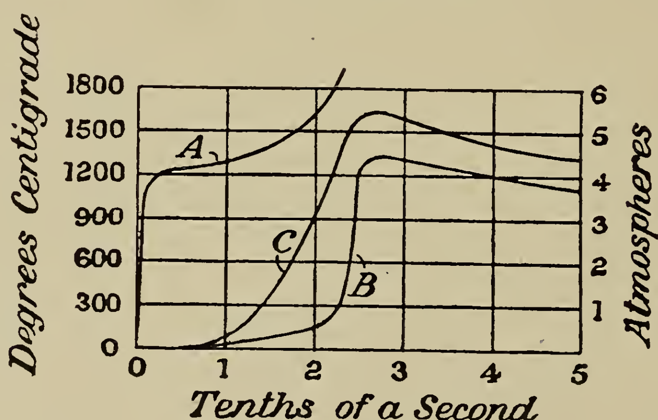
The following table gives the values of specific heat and volumetric heat for the temperatures named.

Temperature	Cv	Volumetric heats
Deg. Cent.		Ft.-lb. per standard cu. ft.
250	0·19	21
750	0·23	25
1250	0·27	29
1750	0·30	33

Although these values refer to the expanding gases in a gas engine, they may also be applied with approximately correct results to the gaseous mixture before explosion. Using this table it is possible therefore to estimate temperature rises corresponding to various amounts of heat energy supplied. The temperatures so estimated will, of course, only be approximately correct, unless the temperature range happened to be centred around one of the above temperatures. It is interesting to compare this table with that on p. 61.

EXAMPLE

1. A mixture of coal-gas and air containing 10 per cent. of coal-gas is fired in a large spherical vessel by a spark at the centre, and the temperature of the gas is recorded by platinum thermometers, one of which



(A) is placed close to the spark and the other (B) near to the walls of the vessel. The records of temperature (curves A and B) and the simultaneous record of pressure (curve C) are shown in the figure. The

records start from the moment of firing. Explain the characteristic features of the temperature records, particularly the rise which occurs in A after 0·1 secs. and the slow rise in B during the first two-tenths of a second.

Estimate from record A the volumetric heat of the products of combustion over the range 1200° to 1800° C.

[Mech. Sc. Tripos, 1913.]

CHAPTER IV

Thermodynamics

Internal Energy—Joule's Law of Thermodynamics—Effect of increasing Specific Heat—Form of Adiabatic—Measurement of Cylinder Temperature—Gas Standard of Efficiency—Flow of Heat through Metal Walls of Cylinder—Heat Paths.

55. Internal Energy.—The applications of thermodynamics to the study of gas engine problems are numerous and varied. The earlier chapter on the efficiency of cycles of operation will have afforded illustration of this, but it is proposed now to devote further attention to the matter.

The relations between P, V and T of unit weight of a perfect gas have been given as $\frac{PV}{T}=R$; and in thermodynamic calculations it is generally necessary to assume that all gases follow this law, which it happens fortunately they very nearly do. Specific heat has been defined, and it has been shown that the R of the equation above is J $(C_p—C_v)$.

We now return to the consideration of the internal energy of a gas, referred to in par. 24. The internal energy of a gas means the total energy, in ft.-lb., actually in the substance at any instant; to define it absolutely, it is necessary to fix upon a definite state of the gas as the zero state, from which to measure (usually 100° C. is selected as the starting point). Generally, however, we are only concerned with changes in the internal energy. If we denote the internal energy by E, we know that an increase ΔE, due to the reception of ΔH heat units while ΔW ft.-lb. of mechanical work has been done by the gas, is given by

$$\Delta E=J\,.\,\Delta H-\Delta W.$$

56. Joule's law of thermodynamics is that in a perfect gas E depends upon the temperature only: or as it may be more generally stated, E is always the same when the gas returns to the same state.

From Joule's law it follows that—unit weight of gas being taken—

$$\Delta E = JC_v \, . \, \Delta T$$

so that

$$J C_v \, . \, \Delta T = \Delta E = J \, . \, \Delta H - \Delta W = J \, . \, \Delta H - P \, . \, \Delta V$$

(an equation which was established in par. 29).

Thus

$$J \frac{\Delta H}{\Delta V} = JC_v \frac{\Delta T}{\Delta V} + P$$

or in the limiting case

$$J \frac{dH}{dV} = J C_v \frac{dT}{dV} + P \, \ldots \ldots \quad (1)$$

The differential coefficient $\frac{dH}{dV}$, which it is important to note is of the dimensions of a pressure, is an important quantity in gas engine expansion and compression curves, and it is necessary to find an expression for it which can be more quickly dealt with than the above equation (1).

Since
$$\frac{PV}{T} = R$$

$$T = \frac{PV}{R}$$

$$\frac{dT}{dV} = \frac{1}{R}\left(P + V\frac{dP}{dV}\right)$$

$$R = J(C_p - C_v) = J \, C_v(\gamma - 1), \text{ since } \frac{C_p}{C_v} = \gamma$$

$$\text{Therefore } \frac{dT}{dV} = \frac{1}{J \, C_v(\gamma - 1)} \left(P + V \frac{dP}{dV}\right)$$

Combining this with equation (1)

$$J\frac{dH}{dV}=\frac{1}{\gamma-1}\left(P+V\ \frac{dP}{dV}\right)+P$$

$$=\frac{1}{\gamma-1}\left(\gamma\ P+V\frac{dP}{dV}\right)\ \ldots\ldots \quad (2)$$

This equation is often quoted with the J on the left hand side omitted, the H then is supposed to be given in energy units (ft.-lb.).

57. The following table, part of which was calculated from an old low compression gas engine indicator card for Professor Perry's book on the *Steam Engine*, affords an illustration of the use of the formula (2)—γ being there taken as 1·385.

	V	P	$\frac{\Delta P}{\Delta V}$	Average V	Average P	$\frac{dH}{dV}$
Compression	25	14·7				
			−0·96	22·5	17·1	5·4
	20	19·5				
			−1·70	17	24·6	13·4
	14	29·7				
			−3·88	12	37·5	14·0
	10	45·2				
Explosion and Expansion	10	45·2				
			173	10·1	62·4	4,760
	10·2	79·7				
			218	10·3	101·5	6,210
	10·4	123·2				
			173	10·5	140·4	5,230
	10·6	157·7				
			120	10·7	169·7	3,930
	10·8	181·7				
			33	10·9	184·9	1,590
	11·0	188·2				
			−22	11·5	177·2	−19·7
	12·0	166·2				
			−20	12·5	156·2	−87·4
	13	146·2				
			−14·8	14	131·5	−64·9
	15	116·7				
			−10·5	16	106·2	−54·3
	17	95·7				
			−7·5	18	88·2	−33·2
	19	80·7				
			−6·0	20	74·7	−42·8
	21	68·7				
			−5·0	22	63·7	−56·6
	23	58·7				

These figures are plotted to scale in Fig. 22. It will be noted from the table that during compression $\frac{dH}{dV}$ is positive in every case, showing that dH and dV must be of the same sign. As V is decreasing during compression dV must be negative

and therefore dH also. So that during compression the gas is losing heat to the colder walls of the cylinder. The ratio of loss is not great, however, and such as it is it represents the differential effect of the cooling of the walls and the heating by contact with and radiation from the hot piston. During explosion the gas is seen, both from the table and the curve,

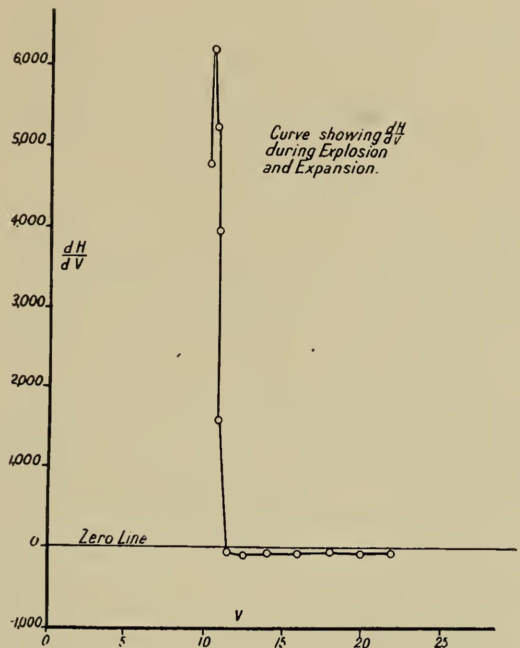


FIG. 22.—Curve of $\frac{dH}{dV}$ and V. It shows how the working stuff receives heat during explosive combustion and how it afterwards loses heat to the walls. Unit of V is arbitrary so unit of $\frac{dH}{dV}$ is arbitrary also.

to gain heat rapidly until the point of greatest pressure and temperature is reached, and then the curve falls rapidly, and the gas begins to show a loss of heat to the cooling walls. This loss has of course been going on during the explosion also, but the effect is masked by the far greater quantities of heat then being liberated. In modern engines, pressures and temperatures are higher, so that increase of specific heat affects the

calculation and must not be neglected. This correction is dealt with later on in the chapter.

58. It is often found that during compression or expansion the gas will follow the law

$$PV^n = \text{constant} = c, \text{ say}$$

so that

$$\frac{dP}{dV} = -\frac{nc}{V^{n+1}}$$

$$= -nPV^n \times \frac{1}{V^{n+1}} = -\frac{nP}{V}$$

or

$$V\frac{dP}{dV} = -nP.$$

Substitute this in equation (2)

and

$$J\frac{dH}{dV} = \frac{1}{\gamma - 1}\left\{-nP + \gamma P\right\} = \frac{\gamma - n}{\gamma - 1}P \quad . \; . \; . \qquad (3)$$

A very simple expression which can often be used to obtain results speedily. If the gas lose in H during compression evidently $(\gamma - n)$ must be positive or γ must be greater than n. During expansion, if the gas is losing heat $(\gamma - n)$ must be negative or n be greater than γ. (Cf. Ex. 14 on p. 40.)

This analysis was originally due to Professors Ayrton and Perry, and published by them in the *Proceedings of the Physical Society* in 1885.

59. Effect of Increasing Specific Heat.—It is important to examine how this calculation is affected when allowance is made for a specific heat which increases with temperature. It was mentioned at the close of last chapter that the gaseous mixtures used in practice had a specific heat value rising in a linear relationship with temperature—in fact that

$$C_v = \beta + sT$$

where β and s were some constants.

Since $(C_p - C_v)$ must from Joule's law be independent of temperature, it follows that

$$C_p = a + sT$$

where s has the same value as above, and a is a new constant. The ratio $\frac{a}{\beta}$ may for convenience be written c; it is obviously the value of γ when $T=0$.

It has just been shown (p. 67) that for unit weight of gas

$$J\frac{dH}{dV}=J\,C_v\,\frac{dT}{dV}+P$$

$$\text{Also that } \frac{dT}{dV}=\frac{1}{R}\left(P+V\frac{dP}{dV}\right)$$

$$\text{but } R=J\,(C_p-C_v)=J(a-\beta)$$

therefore

$$\frac{dT}{dV}=\frac{1}{J\,(a-\beta)}\left(P+V\frac{dP}{dV}\right)$$

so that

$$J\frac{dH}{dV}=\frac{\beta+sT}{a-\beta}\left(P+V\frac{dP}{dV}\right)+P$$

$$=P\left\{1+\frac{\beta+sT}{a-\beta}\right\}+V\frac{dP}{dV}\;\frac{\beta+sT}{a-\beta}$$

$$=\frac{1}{a-\beta}\left\{aP+sTP+\beta V\frac{dP}{dV}+sTV\frac{dP}{dV}\right\}$$

$$\text{so that } J\frac{dH}{dV}=\frac{1}{c-1}\left\{cP+V\frac{dP}{dV}\right\}+\frac{sT}{a-\beta}\left\{P+V\frac{dP}{dV}\right\}\;\ldots \qquad (4)$$

and this is the new expression for $\frac{dH}{dV}$.

If $s=0$; i.e. if specific heats were constant, equation (4) would clearly at once become equation (2).

As before, take the case where, as in compression and expansion, $PV^n=$ constant, very nearly.

Then

$$V\frac{dP}{dV}=-nP,$$

and substituting in equation (4)

$$\mathrm{J}\frac{d\mathrm{H}}{d\mathrm{V}}=\frac{c-n}{c-1}\mathrm{P}+\frac{s\mathrm{T}}{a-\beta}\left\{\mathrm{P}-n\mathrm{P}\right\}$$

or

$$\mathrm{J}\frac{d\mathrm{H}}{d\mathrm{V}}=\left\{\frac{c-n}{c-1}-\frac{n-1}{a-\beta}s\mathrm{T}\right\}\mathrm{P} \quad \ldots\ldots \quad (5)$$

which becomes equal to equation (3) if $s=0$.

Equation (5) shows that $\frac{d\mathrm{H}}{d\mathrm{V}}$ is proportional to P when T is constant, and that it is a linear function of T when P is constant; provided always that all changes are regulated by the law $\mathrm{PV}^n=$ constant.

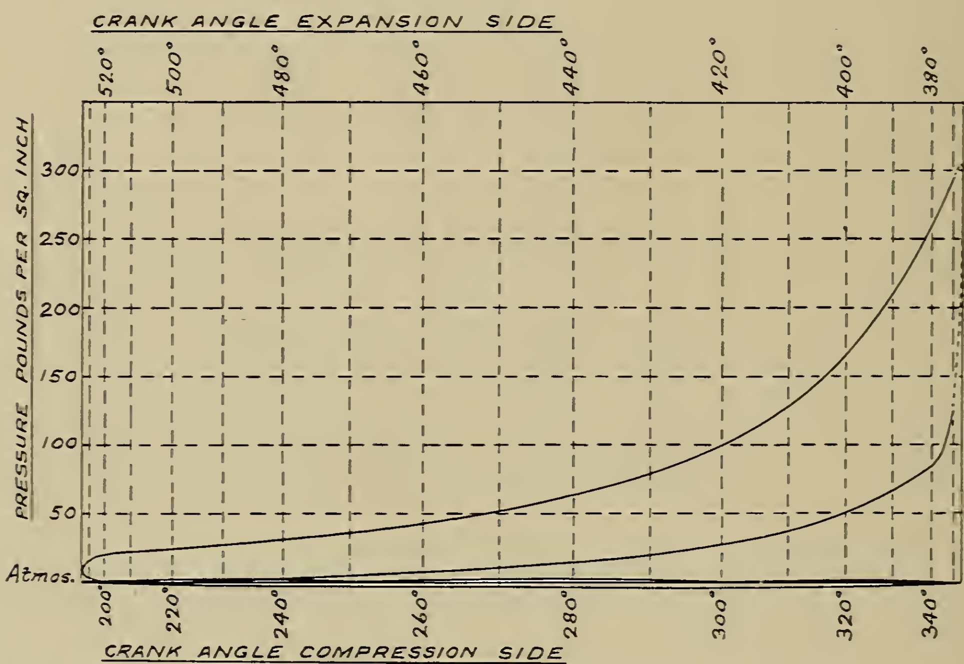


FIG. 23.—Indicator diagram analysed in Fig. 23A. (It is Fig. 5 of the 7th Report of G.E.C.)

Mr. Hogg has applied this method of analysis to an indicator diagram taken with a reflecting indicator by Prof. Dalby and reproduced in Fig. 23. The following are his results.

Value of n during compression 1·372; during expansion 1·435. Composition of gas sensibly the same as that adopted as standard by G.E.C. whose specific heat figures are therefore

taken. Correction for chemical contraction on explosion 2·24 per cent.

Compression				Expansion			
Crank Angle	P (absolute) lb. per sq. inch	$\theta°$ C	$\frac{JdH}{dV}$ ft.-lb. per cub. ft. per lb. of gas	Crank Angle	P (absolute) lb. per sq. inch	$\theta°$ C	$\frac{JdH}{dV}$ ft-.lb. per cub. ft. per lb. of gas
350°	139·8	474	−1610	370°	317·5	1,462	−32,600
340°	102·5	327·6	−333	380°	281	1,405	−27,800
330°	84·0	292	−105	390°	230	1,308	−21,300
320°	67·4	252·7	−57·2	400°	184	1,194	−15,670
310°	54·3	221	+149	410°	152	1,142	−12,420
300°	43·0	183	+211	420°	119	1,017	−8,750
290°	36·2	169·5	+204	430°	96·0	926	−6,490
280°	30·3	149	+205	440°	80·5	872	−5,150
270°	26·0	132	+201	450°	68·3	816	−4,120
260°	—	—	—	460°	59·6	781	−3,450
250°	—	—	—	470°	51·8	732·5	−2,840
240°	—	—	—	480°	47·9	729·4	−2,620
230°	—	—	—	490°	43·0	686	−2,230
220°	—	—	—	500°	40·1	667·8	−2,030
210°	—	—	—	510°	37·7	646·9	−1,855
200°	14·3	77	+154·5	520°	34·2	582·9	−1,540
195°	—	—	—	530°	—	—	—

These figures are plotted in Fig. 23A, and they show the gas to be just losing heat on balance from A to B on the compression stroke, and to be gaining it from B to C. Ignition occurs about the point C, and the curve would then shoot up far off the diagram; this rapid rise cannot be got from the indicator diagram as the volume alters so exceedingly slowly at the dead centre. From this height, however, the curve rapidly descends owing to radiation and convection losses until the point D is reached. The rate of cooling then tends to decrease as shown by the line DE, which represents the expansion period. The gas is still losing heat at the point E, when the exhaust opens and the temperature is in the neighbourhood of 600° C.

Mr. Hogg's curve shows no evidence of any continuation of combustion after the highest temperature has been reached.

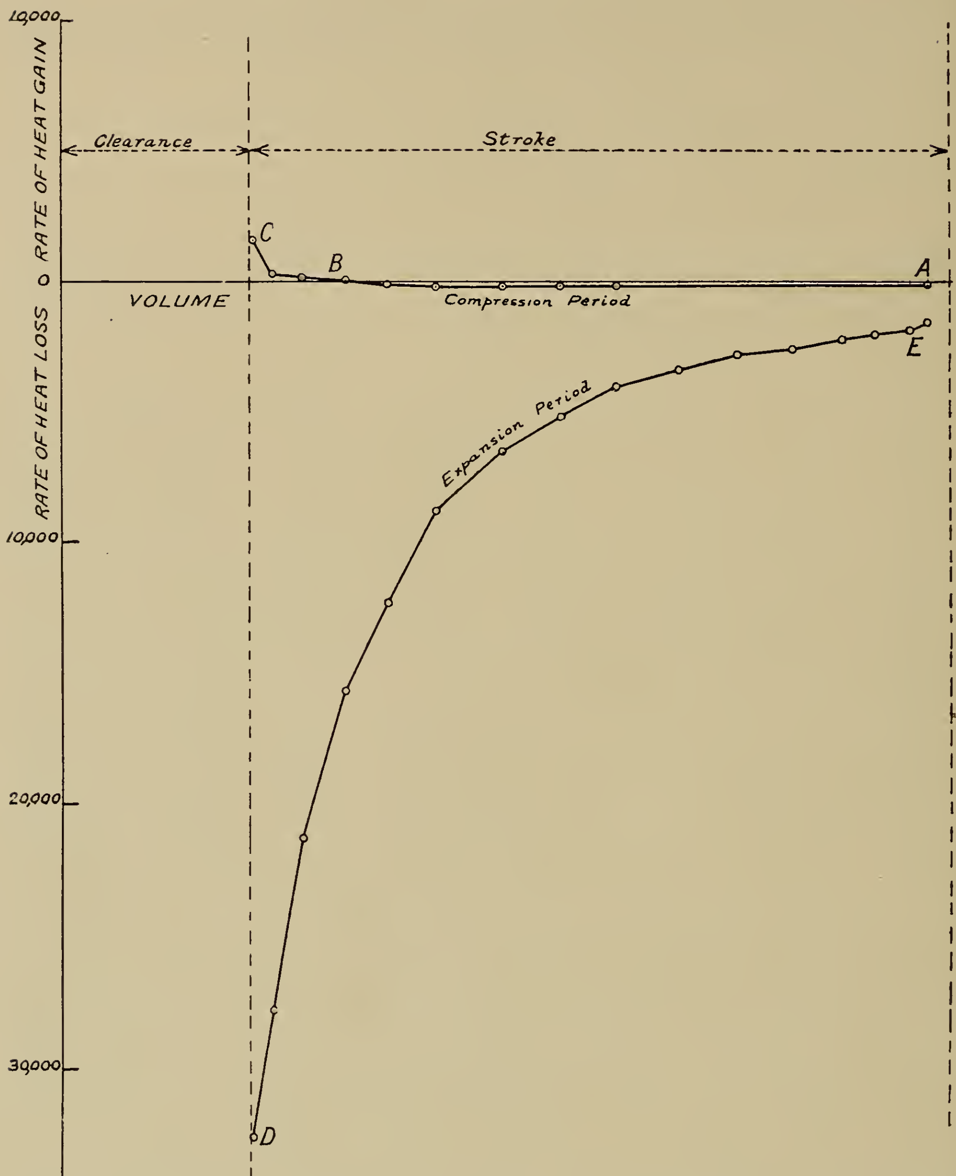


FIG. 23A. Change of Internal Energy during compression and expansion strokes of diagram shown in Fig. 23.

60. Adiabatic law with variable Specific Heats.—If $\frac{dH}{dV}$ be zero, or, in other words, if the transformation be adiabatic, it follows from equation (4) that

$$aP+sTP+\beta V\frac{dP}{dV}+sTV\frac{dP}{dV}=0$$

or

$$V\frac{dP}{dV}+\frac{a+sT}{\beta+sT}P=0.$$

If $s=0$ this would become

$$V\frac{dP}{dV}+\gamma P=0$$

which integrates as in par. 29 to the familiar form $PV^\gamma=$ constant. If, however, s is not zero we have a much harder integration. The above equation becomes

$$\frac{dP}{P}+\frac{a+sT}{\beta+sT}\frac{dV}{V}=0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

Also since $\frac{PV}{T}=\text{constant}$

$$\frac{P}{T}dV+\frac{V}{T}dP-\frac{PV}{T^2}dT=0$$

$$\text{or } \frac{dV}{V}+\frac{dP}{P}-\frac{dT}{T}=0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

Equation (1) may be written

$$\beta\frac{dP}{P}+a\frac{dV}{V}+sT\left(\frac{dP}{P}+\frac{dV}{V}\right)=0$$

or using (2)

$$\beta\frac{dP}{P}+a\frac{dV}{V}+sdT=0$$

The integral of which is

$$\beta\log P+a\log V+sT=\text{constant}$$

and this may also be written

$$P^\beta.\ V^a.\ e^{sT}=\text{constant.}$$

This is therefore the adiabatic law with variable specific heats.

61. Experiments on Measuring Temperatures during the Cycle of Operations in a Gas Engine.—Professor Burstall * was the first to do this. He came to the conclusion that with a platinum thermometer it was impossible, owing to the fusing of the fine platinum wire before a sufficient number of observations had been taken, to make such measurements with an engine working on full load. He had therefore to experiment on an engine running light and firing but once in each twelve revolutions. The principle upon which a plat-

* *Phil. Mag.*, 1895, and *Proc. I.M.E.*, 1901.

inum thermometer works is that since the electrical resistance increases with the temperature in accordance with a known law, to measure the resistance of the wire is to measure its temperature at the moment. Professors Callendar and Dalby * have since made additional tests in this direction. These experimenters realized that they could not get a wire which would "stand up" to the temperature of explosion unless it was so thick that it must fail to follow the fluctuating temperatures of the gas with sufficient rapidity. They therefore decided so to arrange the apparatus that they could withdraw the fine thermometric wire from the action of the gases

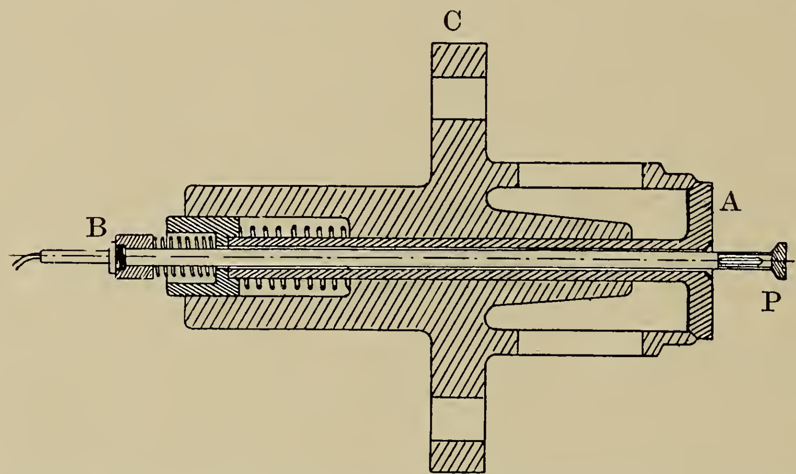


FIG. 24.—Combined Admission and Thermometer Valve (Callendar).

during explosion, and replace it for each suction and compression stroke. This was effected by fitting up the inlet valve as shown in Fig. 24. C is the admission valve casting, which is bolted on to the cylinder and projects inside the space provided for it. The thermometer was inserted through the spindle of the main admission valve marked A, which had been drilled out to receive it. In the figure the little "thermometer valve," as it may be called, is shown projecting beyond the main valve head into the cylinder. It closes with a little conical seating of its own as soon as the ignition point gets near. The thermometer leads enter through B, pass along the thermometer valve spindle until they arrive at the fine platinum wire which is shown at P. The head of the ther-

* *Royal Soc. Proc.*, 1907.

mometer valve is connected to its spindle by the two ribs which are made as thin as possible so that the platinum wire is not screened more than can be helped from the action of the hot gases when the thermometer valve is pushed out into action. The opening and shutting of the thermometer valve at the proper times is effected by suitable mechanism. The thickness of the platinum wire was $\frac{1}{1000}$ of an inch. At about 130 R.P.M. the lag of the thermometer was not more than 10° of crank angle with a temperature fluctuation of nearly 200° in half a revolution. This would correspond to a time lag of $\frac{10}{360} \times \frac{60}{130} = 0{\cdot}013$ sec., which was quite good enough for measuring anything so relatively steady as the suction temperature. As a result of such measurements it was found that the **suction temperature** varied with the conditions of running from about 95° C. at light load to about 125° C. at full load, the air temperature being about 20° C. and the jacket temperature 27° C. The following are details of two tests—

	Test I.	Test II.
R.P.M.	130	114
Ratio air/gas	7·1	5·8
Atmospheric temperature	20° C.	21° C.
Jacket temperature	27° C.	27° C.
Temperature of thermometer valve at 360° crank angle	122° C.	—
Ditto at 26° crank angle	111° C.	130° C.
Corresponding pressure at ditto . . .	18·5 lb./in.^{2}	17·8 lb./in.^{2}
Molecular contraction on combustion .	4·3 per cent.	5·1 per cent.

It was noted that by a curious coincidence the indicator cards from these two trials showed a practically identical expansion curve, not varying by more than 1 lb./in.^{2} at any point. The temperatures during expansion were however far greater when using the richer mixture, and the heat losses to the walls correspondingly greater, so that although much more gas is used in one case than in the other, no more H.P. is obtained, the excess heat units going to waste.

These experiments show that the convenient practice of assuming the suction temperature to be 100° C. irrespective

of load is only approximately correct. When, as in the above experiments, the suction pressure is accurately measured, it is possible to calculate accurately the temperatures throughout the cycle from the perfect gas law and a knowledge of the molecular contraction on combustion.

62. Later Experiments.—Later attempts to measure the temperature of the gas are due to Coker and Scoble. They found the suction temperature to range from 65° C. to 200° C., but there may have been special circumstances causing the range to be wider than usual; their chief work, however, is the measurement of temperature along the expansion curve. To achieve this much more difficult measurement use was made of thermo-couples made up of alloys of platinum with rhodium and iridium, which have very high fusing points, rolled into strips about 5 to 8 ten-thousandths of an inch thick. In this way it was found possible to measure the gas temperature not only on the rising compression curve, but also along the greater part of the expansion curve; the very highest temperatures had still to be estimated, however, by regarding the charge itself as a gas thermometer. The highest temperature points could be more accurately estimated in this way than by inferring them from the suction temperature, since (1) it was not necessary to allow for chemical contraction, (2) and the perfect gas law is more truly followed at high temperatures. It was thus found that the peak temperature in the gas is in the neighbourhood of 1800° C., and may with specially rich mixtures even reach 2000° C.

63. Gas Standard of Efficiency.—In Chap. II the "air standard" of efficiency for internal combustion engines was explained fully, and numerical results were given. The standard so set is, however, much above what could be achieved in any real engine even were *every* source of loss removed. The reason for this is that the gaseous mixture employed in practice, although mainly air, is by no means entirely so, and that in consequence it is incorrect to assume (as is done in the calculation of "air standard" efficiencies) that the specific heat is constant.

It is therefore necessary to make allowance for this, and one way of doing so is to replace the "air standard" with a

"gas standard" based on the mixture commonly used in a gas engine. The best mixture to use is that for which the Gaseous Explosions Committee determined the specific heat-temperature relationship.

As stated on p. 63, this curve corresponds to a linear law between specific heat and temperature agreeing with the observed results within the limits of experimental error, as follows:

$$C_v = 0{\cdot}152 + 0{\cdot}000075\ T,$$

where T is the temp. absolute. This mixture is stated in the Report to be "the mixture on which Clerk experimented." Reference to Clerk's paper (*Proc. Royal Society, A.*, *Vol.* 77) gives the following particulars of this mixture :—

Extreme compositions—

	Vols.		Vols.
Steam (assumed gaseous) . . .	11·2	and	12·7
Carbonic anhydride . . .	4·8	,,	5·5
Oxygen.	8·7	,,	7·0
Nitrogen	75·3	,,	74·8
	100·0		100·0

Corresponding respectively to explosive mixtures containing before combustion 1 volume of gas to 9·8 volumes of air, and 1 volume of gas to 8·5 volumes of air. The lower heat value of the gas was 574 B.Th.U. (or 319 pound-calories) per standard cu. ft., whilst the compression volume was 18·59 per cent. of the total volume. Clerk mentions on p. 334 of Vol. I of his *Gas, Petrol and Oil Engine* that a standard cubic foot of this mixture weighs 0·07833 lb. So that if the specific heat be expressed as ft.-lb. per standard cu. ft. (i.e. as volumetric heat) it becomes

$$= 16{\cdot}6 + 0{\cdot}0082\ T.$$

With this data it is possible to calculate the temperature at any point in the ideal constant volume cycle (see Fig. 7, the lettering of which is also followed in what follows), provided that the suction temperature be known. This may conveni-

ently be taken as 100° C., a figure never far from its actual value, and one moreover which agrees with the use of that temperature as a starting point for internal energy measurements. We may assume for simplicity's sake that our ideal engine cylinder when filled at the end of the suction stroke contains exactly one standard cu. ft. of gaseous mixture made up of a small proportion of exhaust products together with fresh air and fresh gas. The amount of the exhaust products will vary with the compression ratio, but no great error will be made by taking it that at the end of the suction stroke the contents of the cylinder (at 100° C.) are such as to have an average calorific value of 29 pound calories, being made up of, say, 1 part by volume of gas to 10 parts by volume of a combination of air and exhaust products, so that the volume of gas present is one-eleventh part of a standard cubic foot containing 319 ÷ 11 or 29 pound calories of heat energy.

64. The Adiabatic Curves of Compression and Expansion (var. sp. heats).—The theoretical efficiency of the new standard cycle can now be worked out for various compression ratios, assuming no heat loss, no chemical change of volume, no change on explosion of the relationship between specific heat and temperature, no combustion after the point of maximum temperature, and the suction temperature constant at 100° C. Knowing the suction temperature (T_0), the temperature at the end of compression (T_1) can be calculated from the adiabatic formula appropriate to a linear relationship between specific heat and temperature. Having the compression temperature (T_1) it is easy to obtain the explosion temperature (T_2) from the heat liberated (29 pound calories), and the known specific heat values. The temperature at the end of expansion (T_3) is obtained from the same law as that governing compression. All four temperatures being then known, and the internal energy corresponding to each, the calculation of efficiency follows at once.

The adiabatic law corresponding to a linear relationship between specific heat and temperature is as given in par. 60 $P^{\beta} \, . \, V^{\alpha} \, . \, e^{sT} =$ constant.

$$\text{But } \frac{PV}{T} = \text{constant.}$$

Therefore $V^{\alpha-\beta} . e^{sT} . T^{\beta}=\text{constant}.$

Or
$$\left(\frac{V_0}{V_1}\right)^{\alpha-\beta}\left(\frac{T_0}{T_1}\right)^{\beta} . e^{s(T_0-T_1)}=1.$$
$$\left(\frac{V_0}{V_1}\right)^{\alpha-\beta}=\left(\frac{T_1}{T_0}\right)^{\beta} . e^{s(T_1-T_0)}$$
$$\therefore \left(\frac{V_0}{V_1}\right)=\left(\frac{T_1}{T_0}\right)^{\frac{\beta}{\alpha-\beta}} . e^{\frac{s}{\alpha-\beta}(T_1-T_0)}$$

It is not possible to calculate T_1 directly from this, but curves may be drawn giving the relationship of temperature and compression ratio, and in this way T_1 for any given volume ratio (r) may be found.

Now $T_0 = 373$, and $C_v=\beta+sT=0{\cdot}152+0{\cdot}000075\ T$.

To get C_p we need R.

Now $C_p - C_v = \frac{R}{J}$ and $R=\frac{PV}{T}=\frac{14{\cdot}7\times144}{0{\cdot}07833\times273}=99$; and $C_p-C_v=0{\cdot}071$.

So that $C_p=\alpha+sT=0{\cdot}223+0{\cdot}000075\ T$.

Therefore
$$\log_\epsilon\left(\frac{V_0}{V_1}\right)=2{\cdot}14\ \log_\epsilon\left(\frac{T_1}{373}\right)+\frac{T_1-373}{950}=\log_\epsilon r.$$

The relationship of T_1 and r are shown in the Table below.

θ_1	T_1	r	θ_1 (continued)	T_1 (continued)	r (continued)
100	373	1	1,100	1,373	46·5
150	423	1·38	1,200	1,473	60·1
200	473	1·85	1,300	1,573	77·1
250	523	2·39	1,400	1,673	97·3
300	573	3·09	1,500	1,773	122
350	623	3·89	1,600	1,873	153
400	673	4·86	1,700	1,973	190
450	723	5·97	1,800	2,073	235
500	773	7·24	1,900	2,173	289
600	873	10·4	2,000	2,273	353
700	973	14·7	2,100	2,373	430
800	1,073	20·1	2,200	2,473	522
900	1,173	26·9	2,300	2,573	631
1,000	1,273	35·6			

By plotting the values of T_1 and r from this Table, it is possible to deduce the values of T_1 for specific values of the compression ratio, as shown below.

CORRESPONDING VALUES OF r and TEMP. (ABS.) AT END OF COMPRESSION.

r	T_1	r	T_1
1	373	9	831
2	487	10	861
3	567	11	887
4	629	12	914
5	680	13	937
6	725	14	959
7	763	15	981
8	799		

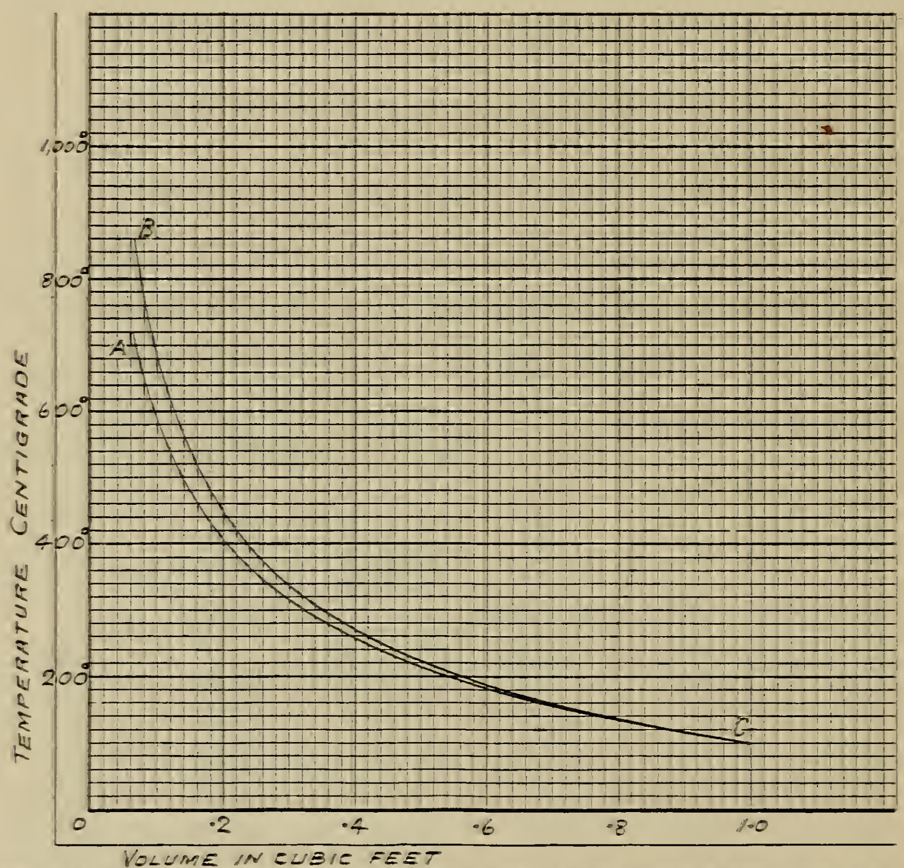


FIG. 24A.—Compression curves for one cubic foot of air originally at 100° C. CB with constant specific heat ($PV^{1 \cdot 41}$ = const.); CA with variable specific heat (gas engine mixture).

It is useful to express these temperatures in other than the absolute form, and to compare them with the corresponding temperatures obtained from the adiabatic law for a perfect gas with constant specific heat, viz. $PV^{1.41} =$ constant. This comparison is given below, and is illustrated in Fig. 24A.

r	θ° C. (var. specific heat)	θ° C. (const. specific heat)
1	100	100
2	214	223
3	294	312
4	356	386
5	407	449
6	452	505
7	490	555
8	526	602
9	558	645
10	588	686
11	614	724
12	641	760
13	664	795
14	686	828
15	708	859

65. Internal Energy Throughout Cycle.—Having thus found T_1 for various values of r, it is next necessary to obtain T_2 the explosion temperature, and this is best got from an internal energy curve.

$$\text{Volumetric heat} = 16{\cdot}6 + 0{\cdot}0082\ T$$

and the internal energy

$$= \int_{373}^{T} (16{\cdot}6 + 0{\cdot}0082\ T)\,dT$$

$$= 16{\cdot}6\ (T - 373) + 0{\cdot}0041(T^2 - 373^2)\ \text{ft.-lb.}$$

This is shown plotted in Fig. 25. Now on explosion, 29

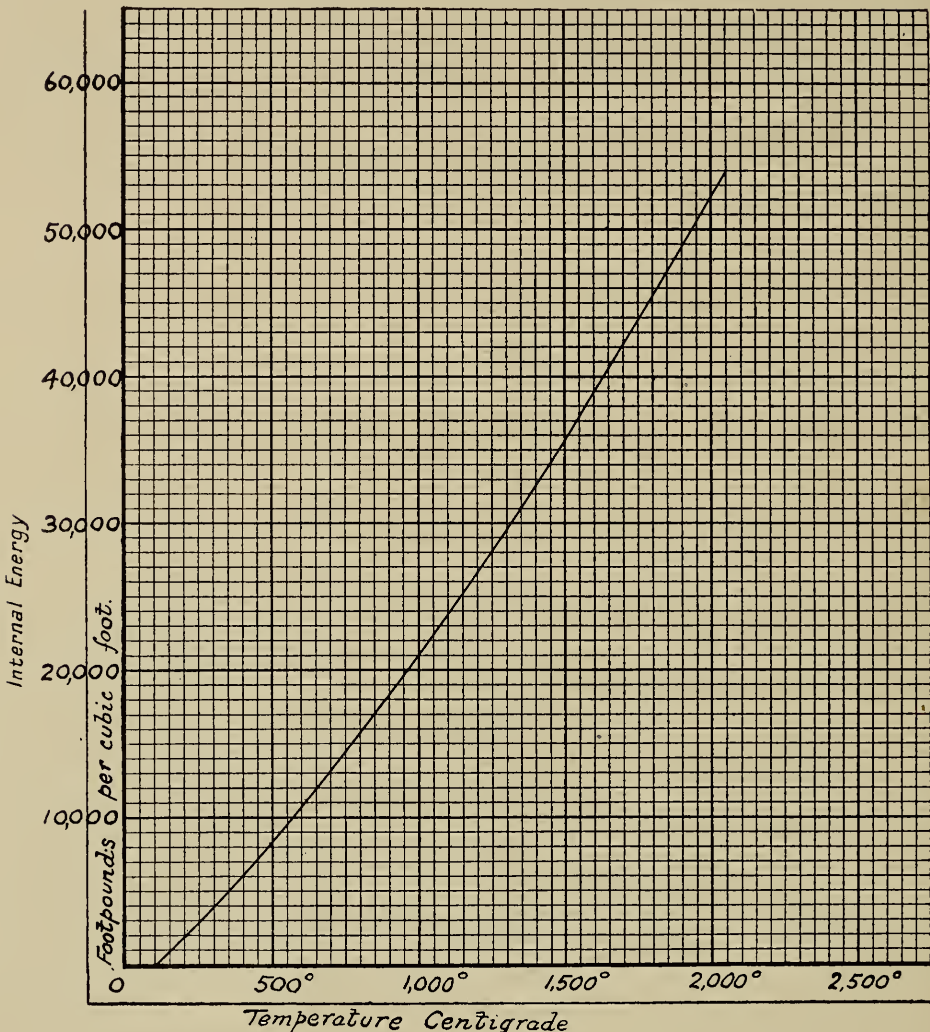


FIG. 25.—Internal Energy—Temperature graph for Gas Engine Mixture (G.E.C.).

pound-calories of energy are given to the gas, equal to 29 × 1400 = 40,600 ft.-lb. The values of the energy for various values of the temperature can conveniently be plotted from the above equation, and if to the energy value at any particular compression temperature 40,600 ft.-lb. be added, the total is the energy figure for the corresponding explosion temperature. Then from the same energy curve the explosion temperatures can be themselves deduced. These temperatures can be subquently checked by calculation where necessary. The following are the results obtained :—

TABLE OF TEMPERATURES.

r	θ_0 °C. (suction)	θ_1 °C. (compression)	θ_2 °C. (explosion)	θ_3 °C. (exhaust)
1	100	100	1,660	1,660
2	100	214	1,730	1,415
3	100	294	1,780	1,290
4	100	356	1,820	1,205
5	100	407	1,850	1,145
6	100	452	1,880	1,100
7	100	490	1,905	1,060
8	100	526	1,930	1,025
9	100	558	1,950	1,000
10	100	588	1,970	976
11	100	614	1,990	955
12	100	641	2,010	936
13	100	664	2,025	920
14	100	686	2,040	904
15	100	708	2,050	889

TABLE OF ENERGY AND EFFICIENCY.

r	Internal Energy in Ft.-lb.				Thermal Efficiency.		
	E_0 (Suction).	E_1 (Compression).	E_2 (Explosion).	E_3 (Exhaust).	"Gas Standard."	"Air Standard."	Ratio of Gas Standard to Air.
1	0	0	40,600	40,600	0	0	—
2	0	2,300	42,900	33,000	0·188	0·242	·78
3	0	3,950	44,550	29,200	0·281	0·356	·79
4	0	5,300	45,900	26,750	0·341	0·426	·80
5	0	6,450	47,050	25,000	0·384	0·475	·81
6	0	7,450	48,050	23,700	0·417	0·512	·81
7	0	8,300	48,900	22,600	0·443	0·541	·82
8	0	9,100	49,700	21,650	0·467	0·565	·83
9	0	9,900	50,500	21,000	0·483	0·585	·83
10	0	10,550	51,150	20,350	0·498	0·602	·83
11	0	11,200	51,800	19,800	0·512	0·617	·83
12	0	11,850	52,450	19,300	0·524	0·630	·83
13	0	12,400	53,000	18,850	0·536	0·642	·83
14	0	12,950	53,550	18,450	0·545	0·652	·84
15	0	13,350	53,950	18,050	0·555	0·661	·84

66. Gas Standard.—It is now possible to give the value of the internal energy at each of the four corners of the diagram, and so to obtain the thermal efficiency corresponding to each compression ratio, as shown on p. 86.

It will be seen that with this gas mixture the gas standard is about 80 per cent. of the air standard for the compression ratios in common use.* It corresponds very closely to the value $1-\left(\frac{1}{r}\right)^{0\cdot30}$.

67. Approximate Formula for Gas Standard.—It is not difficult to show that to a first approximation the "gas standard" efficiency is given by the expression

$$\eta\left\{1-(1-\eta)\frac{s}{\beta}\,\frac{T_2+T_1}{2}\right\}$$

where $\eta=1-\left(\frac{1}{r}\right)^{c-1}$.

Or if it be preferred, the following approximation may be used

$$\eta\left\{1-\frac{s}{2\beta}\overline{(1-\eta}\,.\,T_2+T_0)\right\}.$$

It is a useful exercise for students to compare this approximate rule with the figures of p. 85.

68. Hopkinson's Efficiency Experiments.—In a paper presented to the Institution of Mechanical Engineers in 1908, Prof. Hopkinson described certain experiments he had made to determine the relationship of actual engine efficiency with the "air standard" and with what we have termed the "gas standard." His results are conveniently summarized in Fig. 26. The uppermost dotted curve is the "Air Standard" which for the compression selected (viz. $r = 6\cdot37$) comes out at 52·2 per cent. Under that is a line which was calculated by Hopkinson on the basis of a variable specific heat (using the figures of Holborn and Austin, and Langen). Below that again is the line of efficiencies as actually found.

* Prof. Asakawa (Brit. Association, 1913) has studied experimentally the effect of variation of compression ratio on thermal efficiency. It appears that about 85 per cent. of the "gas standard" efficiency may be expected as "indicated thermal efficiency" in practice.

The second line was calculated by an approximate graphical method.

69. Choice of Working Fluid.—What change would be effected in the thermal efficiency of an engine if the working fluid were changed for one having a **larger specific heat?** This is an important problem, as it not only concerns the choice of working fluid, but also whether it is well to work high up the temperature scale or not (the specific heat increasing with the temperature as has been shown).

The thermal efficiency of an engine depends on many factors, but to a first approximation it may be taken as proportional,

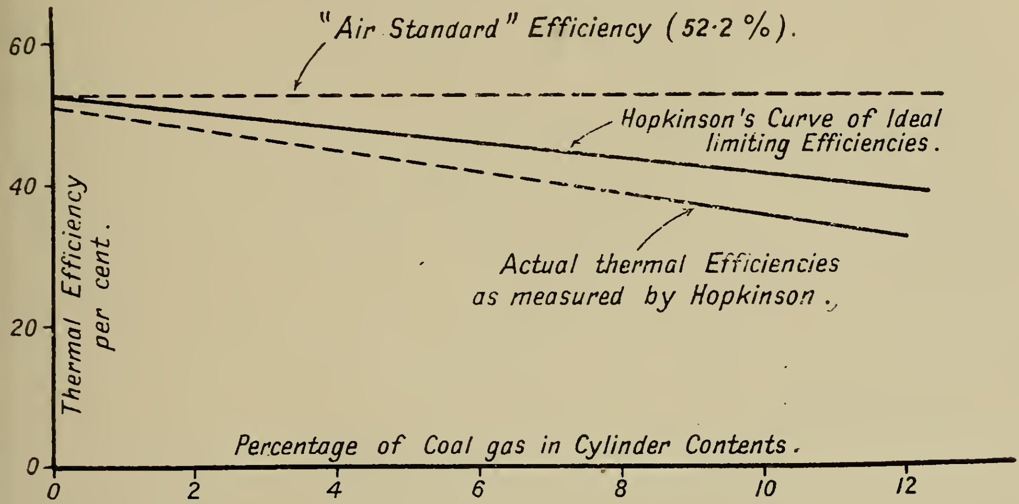


FIG. 26.—Hopkinson's measurements of actual thermal efficiency for mixtures containing from 8 to 12 per cent. of coal gas, compared with his calculated ideal limiting efficiency curve.

for any given compression, to the efficiency as obtained from the "Air Standard" formula

$$\eta=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

Now $$J(C_p-C_v)=R \text{ and } \gamma-1=\frac{R}{JC_v}$$

so that $$\eta=1-\left(\frac{1}{r}\right)^{\frac{R}{JC_v}}$$

$$1-\eta=\left(\frac{1}{r}\right)^{\frac{R}{JC_v}}$$

$$\log(1-\eta)=\frac{R}{JC_v}\log\frac{1}{r}$$

differentiate
$$\frac{-1}{1-\eta}\frac{d\eta}{dC_v}=\frac{R}{JC_v^2}\log r.$$
$$\frac{d\eta}{dC_v}=-\frac{R(1-\eta)}{JC_v^2}\log r.$$

Therefore *with increase of specific heat the efficiency falls.*

This could also be written

$$\left(\frac{d\eta}{\eta}\right)=-\left(\frac{dC_v}{C_v}\right)\left\{\frac{R}{JC_v}\ \frac{1-\eta}{\eta}\log r\right\}$$
$$=-\left(\frac{dC_v}{C_v}\right)\left\{(\gamma-1)\frac{1-\eta}{\eta}\log r\right\}$$

This gives the proportional change in efficiency for a given proportional change in specific heat.

If for example $\gamma = 1{\cdot}40$ and $r = 10$, then for a 1 per cent. increase in C_v the corresponding proportional decrease in efficiency would be

$$\frac{1}{100}\left\{(1{\cdot}4-1)\frac{1-\eta}{\eta}\log_\epsilon 10\right\}$$

Now when $r = 10$, $\eta = 0{\cdot}60$

and
$$\frac{d\eta}{\eta}=\left\{0{\cdot}40\times\frac{0{\cdot}40}{0{\cdot}60}\times 2{\cdot}30\right\}\times\frac{1}{100}$$
$$=0{\cdot}61 \text{ per cent.}$$

So that in this case the efficiency falls by **rather more than ½ per cent.** when the specific heat rises by 1 per cent.

70. Heat Flow through Cylinder Walls.—One of the most important matters connected with the temperature changes in an internal combustion engine is the consideration of the manner in which the heat carried away by the cooling water passes from the hot gas to the water. The cooling water is circulated around outside of the cylinder in a space provided between the cylinder walls and the jacket. The carrying away of heat by this cooling water is, of course, of no thermal advantage to the engine, much the contrary, in fact ; but unless it is allowed to take place the cylinder walls would reach so high

a temperature that lubrication would become impossible. Engines become more efficient as this heat loss is reduced, but care has to be taken to limit the reduction at the point where there would be risk of the lubrication failing. If the lubrication did so fail the piston would seize and the engine be seriously damaged. In a four-stroke engine of the usual type the temperature of the gases inside the cylinder will vary during the cycle from about 20° C. to 1500° C. This is a large range, but it is found that the wall temperature—even that of its innermost face—does not pass through anything approaching so large an amount. With a temperature range in the gas of the amount mentioned the total temperature range in the inner face of the wall will not exceed about 10° C. And even this small range is but skin-deep. It has been shown mathematically, and confirmed by experiment, that at a depth below the inner surface of the wall of $\frac{3}{16}$ inch the temperature range is only about $\frac{1}{250}$th part of the range at the surface. The following comparative statement can therefore be given as an illustration :—

Range of temperature in gas—about 1500° C.
,, ,, ,, in wall surface—about 10° C.
,, ,, ,, $\frac{3}{16}$ inch deep in wall—about $\frac{1}{25}$° C.

For all practical purposes, therefore, the wall temperature as a whole does *not* vary from moment to moment during the cycle.

The temperature is different, however, in different parts of the length of the wall. Thus, the wall is hotter near the compression head than it is at the other end of the cylinder. This is because most of the heat loss occurs at the beginning of the stroke, in the clearance space, where the highest gas temperatures are found. The consequent varying amount of expansion in the different cylinder parts renders the metal liable to crack unless the design provides room for expansion. Cylinder heads and pistons are the most difficult parts to protect against cracking, particularly in large engines. But if fixed joints between metal faces necessarily at different temperatures are replaced by sliding ones, much of the trouble may be removed.

71. Hopkinson's and Coker's Experiments on Cylinder Temperatures.—Experiments made by Hopkinson * and by Coker † on different engines afford a basis for drawing up a table of the probable temperatures occurring in the ordinary type of four-stroke engine (open ended) :—

Part.	Temperature.
Maximum temperature of gas, say	1,500° C.
Piston, centre of face	400° C.
Centre of exhaust valve	400° C.
Ditto, inlet valve	250° C.
Unjacketed part of wall in clearance space .	250° C.
Piston, edge of face	200° C.
Jacketed wall in clearance space	100° C.
Jacketed part of wall in stroke	75° C.
Cooling water at outlet	65° C.

Coker also found that with a copious supply of cooling water there was no difference in wall temperature along the length of the stroke. But with a restricted supply the average wall temperature was less (by some 7° C.) in the metal near the end of the outward stroke than at the beginning. Thus when the cooling water is restricted in quantity some of the heat passing into the cylinder walls travels *along* the wall as well as *through* it.

An interesting measurement made by Hopkinson showed that if any part inside the cylinder should rise to 700° C. the gases would ignite spontaneously, so causing *pre-ignition.*

72. Mathematical Theory of Wall Temperatures.—It is interesting to investigate this problem mathematically.

Let OO be the inner face of the section of the wall which can with sufficient accuracy for this problem be considered plane. Consider what is happening at A, distant x below the surface of the metal. Across an imaginary unit area perpendicular to the surface of the paper and to the line of flow of the heat which is in the direction of the arrow, heat will be transmitted

* *Proc. I.C.E.* 1909.
† *Proc. I.C.E.* 1913.

but a part will be retained for the heating up of the substance of the lamina at A. At a section at distance $(x + \delta x)$ the temperature will be $(\theta + \delta\theta)$, where of course $\delta\theta$ is negative,

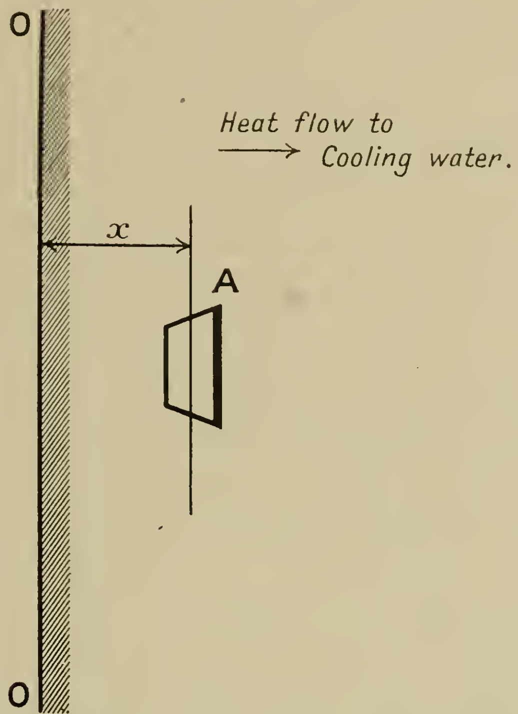


FIG. 27.

at the same moment of time. Now the rate at which heat is received at the left face of the lamina, contained by the two planes at x and $(x + \delta x)$, is equal to $- k \dfrac{d\theta}{dx}$ where k is the conductivity, and the additional amount which flows out per second on the other side is $\dfrac{d}{dx}\left(- k \dfrac{d\theta}{dx}\right) \delta x = - k \dfrac{d^2\theta}{dx^2} \delta x.$ Now this heat must be equal to that required to raise the temperature of the lamina between the time t and the time $(t + \delta t)$, and the volume of the lamina being $(1 \times 1 \times \delta x) = \delta x$, it follows that the heat so absorbed must be equal to $\delta x \, w \dfrac{d\theta}{dt} \delta t \, \sigma$, where w = weight of unit volume and σ = specific heat.

Therefore

$$-k\frac{d^2\theta}{dx^2}\ \delta x\ \delta t = -\ w\sigma\ \frac{d\theta}{dt}\delta x\ \delta t$$

or

$$\frac{k}{w\sigma}\frac{d^2\theta}{dx^2}=\frac{d\theta}{dt}\ \ldots\ldots\ldots\ (1)$$

This is the equation for the flow of heat. The same equation occurs in problems relating to electric conductivity, to the diffusion of liquids into each other and to many other physical applications. Its solution is therefore well known, and in this case the simplest form of it is

$$\theta=\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x)\ \ldots\ldots\ldots\ (2)$$

where C, a, γ and β are constants some of which can immediately be determined from equation (1).

From (2)

$$\frac{d\theta}{dt}=\gamma\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x)$$

$$\frac{d\theta}{dx}=-a\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x)-\beta\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x)$$

$$\frac{d^2\theta}{dx^2}=a^2\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x)+a\beta\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x)+a\beta\mathrm{C}\epsilon^{-ax}\cos$$
$$(\gamma t-\beta x)-\beta^2\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x).$$
$$=(a^2-\beta^2)\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x)+2a\beta\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x)$$

So that equation (1) may be written

$$\frac{k}{w\sigma}\left\{(a^2-\beta^2)\mathrm{C}\epsilon^{-ax}\sin(\gamma t-\beta x)+2a\beta\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x)\right\}$$
$$=\gamma\mathrm{C}\epsilon^{-ax}\cos(\gamma t-\beta x).$$

For this to be an identity

$$a^2-\beta^2=0$$

and

$$2a\beta\frac{k}{w\sigma}=\gamma$$

Now since when $x=a$, the value of θ may be regarded as the zero from which the temperature is measured, it follows that a must be positive and real.

So that $$\alpha=\beta=\sqrt{\frac{w\sigma\gamma}{2k}}$$

Substituting in equation (2)

$$\theta=\mathrm{C}.\epsilon^{-\sqrt{\frac{w\sigma\gamma}{2k}}.x}\sin\left(\gamma t-\sqrt{\frac{w\sigma\gamma}{2k}}.x\right)\quad .\ .\ .\ .\ .\quad (3)$$

73. Skin Temperature.—Now when $x=o$ the value of θ is that for what may be called the skin temperature of the metal—call this θ_0

therefore $$\theta_0=\mathrm{C}\sin\gamma t,$$

This calculation indicates the existence of a skin temperature in the metal which rises and falls as a simple harmonic function of the time with an amplitude of C, that is to say the range of temperature in the skin is 2C. Now imagine the wall to be in contact with a highly heated gas the temperature of which fluctuates rapidly and unevenly. It is well known that by Fourier analysis this temperature can be represented by a series of simple harmonic functions of the time, of increasing frequencies. In a gas engine the temperature of the gas rises and falls about a mean value in what is roughly a sine curve, and in any case the addition of two or three upper harmonics should make the representation very close. The effect of high harmonics at the interior part of the wall is, however, slight; since it will be observed that the logarithmic decrement factor in equation (3) becomes more and more prominent as γ increases in value. It will therefore be sufficiently accurate in the first instance to consider the fundamental period only and to assume that it causes in the skin of the metal a fluctuation of temperature of much the same nature but of less amplitude and with at least some lag. What this amplitude and lag will be it is impossible to calculate, but it is possible to take the extreme case in which the range of temperature in this skin is *equal* to that in the gas. This at least will represent the *limit* of what can occur in that direction. Then $\theta=\mathrm{C}\sin\gamma t$ is the equation for the temperatures both of gas and skin.

74. Effect at a Depth.—It remains to investigate how the rest of the metal wall is affected by this great vibration in temperature in one of its faces. It is clear from equation (3)

that the amplitude decreases with the depth in the metal and that a lag arises and increases at the same time. The amplitude at any point at a depth x is C exp.$\left(-\sqrt{\frac{w\sigma\gamma}{2k}}\,.\,x\right)$, but C is the amplitude at the surface, and therefore the fractional amplitude in the interior is exp. $\left(-\sqrt{\frac{w\sigma\gamma}{2k}}.x\right)$.

It is of interest to evaluate this expression.

We may put $w = 450$ lb. per cubic foot; $\gamma = 10\pi$ corresponding on the basis of a two stroke cycle to a speed of 300 r.p.m.; $\sigma = 0{\cdot}1$; $k = 0{\cdot}01$. So that

$$\frac{w\sigma\gamma}{2k} = \frac{450}{2} \times 0{\cdot}1 \times 10\pi \times 100 = 70{,}500$$

or

$$\sqrt{\frac{w\sigma\gamma}{2k}} = 266$$

and the fractional amplitude $= \epsilon^{-266x}$

here of course x is in feet. If x be put equal to $\frac{1}{4}$ in. or $\frac{1}{48}$ foot, the fractional amplitude $= \frac{1}{\epsilon^{5.53}} = \frac{1}{250}$ or $0{\cdot}40$ of one per cent., which shows that even at a depth of only $\frac{1}{4}$ inch the temperature oscillation is practically wiped out. The curve in Fig. 28 shows graphically how rapidly the oscillations decrease in amplitude. So that in assuming the wall to be infinitely thick no very far-reaching assumption was made, since for anything over $\frac{1}{4}$ in. in thickness the temperature on the water side will practically show no temperature oscillation.

75. Conclusions.—This mathematical investigation shows that the *temperature gradient from face to face of the wall is practically unaffected by the oscillation in the temperature of the gas*, and that if to this gradient line the above shallow temperature oscillations be added a representation can readily be obtained of what is actually occurring in the walls of a gas engine cylinder. The heat flow through the metal is known, as regards quantity, from the heat balance-sheet for the engine, since the heat taken away by the cooling water must be exactly equal to the flow through the walls if a steady state has been reached.

The difference in temperature between outer skin and water must just be enough to enable this amount of heat to pass. In a 10 in. × 18 in. two stroke engine which loses say 30 H.P. continuously through the walls of an exposed area of 4 sq. feet, the average temperature gradient * will be $\frac{7\frac{1}{2}}{3} \times 10 = 25°$ C. per inch. If the inner skin be at an average temperature of 200° C., then the outer skin would be at 175° C.

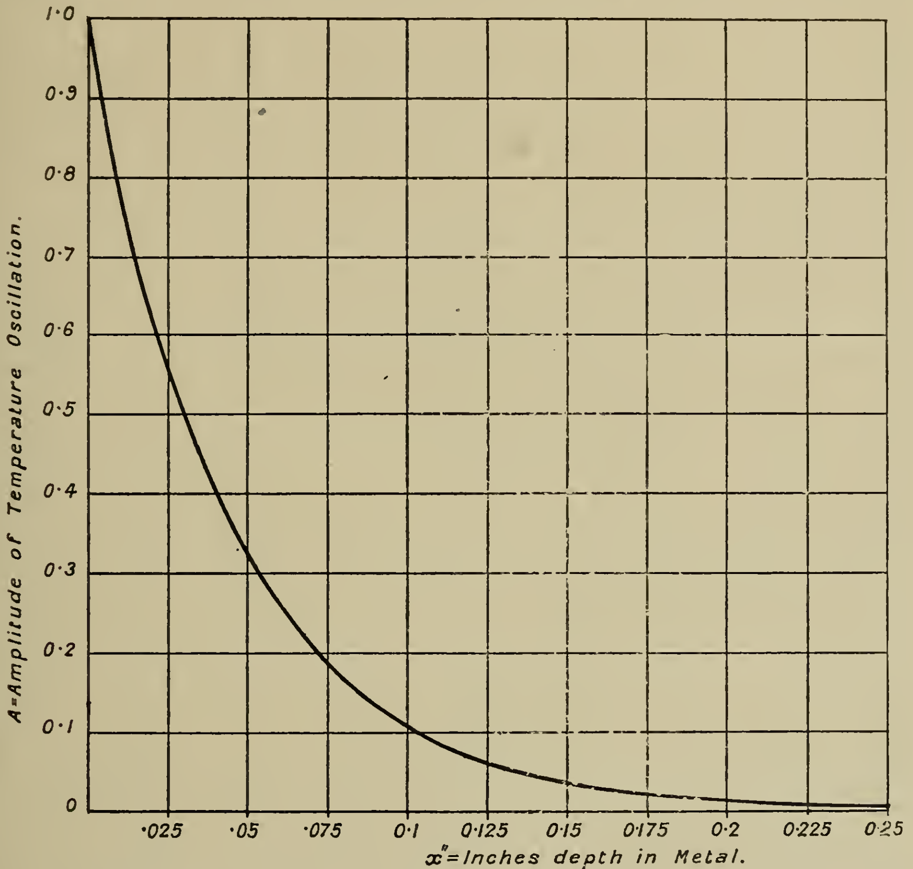


FIG. 28.—Showing amplitude of temperature oscillation in an infinite block of metal when the skin temperature oscillates above and below the mean temperature of the block.

From the equations of par. 72 it is possible to calculate the flow of heat through the inner skin into the metal during the period of time (i.e. the explosion stroke) in which the skin temperature is greater than that of the mass of the metal.

* A plate of average iron, having a temperature gradient of one degree Cent. per inch of thickness, will steadily transmit per square foot of area about $\frac{3}{10}$ horse power.

Thus the heat flow at the surface must be $-k\left(\frac{d\theta}{dx}\right)_{x=0}$

$$=k\,.\,a\,.\,C\,(\sin\gamma t+\cos\gamma t)=k\,.\,a\,.\,C\cdot\sqrt{2}\sin\left(\gamma t+\frac{\pi}{4}\right).$$

The amount of heat flowing per sq. ft. between times $t=o$ and $t=\frac{\pi}{\gamma}$ must be $=\int_0^{\frac{\pi}{\gamma}} kaC\sqrt{2}\sin\left(\gamma t+\frac{\pi}{4}\right).\,dt=\frac{2kaC}{\gamma}$. Inserting the values of the constants of the previous paragraph we have heat flow $=2\,\frac{0{\cdot}01\times266\times C}{10\pi}=0{\cdot}17C$. Take for example a heat loss equivalent to $7\frac{1}{2}$ H.P. per sq. ft. of surface, then heat flow in one cycle (at 300 r.p.m.) $=7{\cdot}5\times\frac{550}{1400}\times0{\cdot}2$ $=0{\cdot}59$ pound-calories, which may be equated to $0{\cdot}17C$, giving $C=3{\cdot}5$ deg., or a total temperature oscillation of 7 degrees. This indicates generally that a small oscillation in the temperature of the innermost layer of metal is quite sufficient to absorb and level the temperature oscillations which the gas tends to set up in the cylinder walls. We may conclude from this that although the temperature at any point of the walls depends on the position of that point, it does not sensibly vary with the time: that is to say, that at any given point the temperature in the walls remains nearly steady. The wall may therefore be considered as of two parts—the inner skin which acts as an accumulator of heat energy, rapidly abstracting it during explosion and giving it out again later; and another part, consisting of the whole of the rest of the wall, which acts as a steady transmitter of the heat fed into it through the inner layer.

76. Heat Paths.—A simple picture can now be drawn of the manner of heat flow in a gas engine cylinder; heat passes to the walls by convection and radiation: this is facilitated by the cooling of the walls by the water circulation. Heat flows most freely into those parts of the metal which are in the explosion space. Valve seatings, ignition bosses, and parts supported by webs tend to get hottest, as the water is more remote and the heat has less chance of escaping. The main

heat flow is straight through the metal of the wall, but it also passes *along* it if the water supply be not copious. A certain amount of heat passes into the piston face. This escapes in two ways—out at the back of the piston, and radially to its edge and along the trunk. All these heat paths require study, so that proper allowances can be made for the expansion of the metal. Ignorance of the way in which this expansion occurs has led to the cracking and failure of a great number of cylinders.

EXAMPLES

1. The temperatures on the two sides of an iron plate 0·5 in. thick differ by 10° C. How much heat in C.H.U. passes through per sq. ft. per minute ? Conductivity of iron is 0·0074 C.H.U. per sq. ft. per sec. for each degree Centigrade drop of temperature per foot of travel.

2. A gaseous mixture has a specific heat at constant pressure of 0·26 and one at constant volume of 0·19. $J = 1393$.

(i) What is the law of adiabatic expansion ?

(ii) If a pound of it is at 120° C., pressure 5000 lb. per sq. foot, what is its volume ?

(iii) A pound of it expands according to the law $PV^s =$ constant ; what is its rate of reception of heat in C.H.U. ?

[B. of E., 1899.]

3. The characteristic equation for the expansion and compression of one pound of air is $PV = 96\ T$,

where P = pressure in lb. per sq. ft.

V = volume in cu. ft. T = temperature abs. in degrees centigrade.

One pound of air expands from an initial pressure of 50 lb. per sq. inch abs. to a pressure of 20 lb. per sq. inch abs., the corresponding change of volume is from 4 to 15 cu. ft., and the mean value of the pressure during the expansion (as deduced from a diagram) is 32·7 lb. per sq. inch.

Calculate—

(i) The initial temperature.

(ii) The final temperature.

(iii) The change in internal energy and the heat received or rejected during the change from the initial to the final condition. $[C_v = 0{\cdot}169.]$

[B. of E., 1912.]

4. Air is compressed adiabatically from 15 to 700 lb. per sq. inch in two stages, the air being cooled finally, and also between the stages to

the initial temperature, which is 18° C. Calculate the interstage pressure which will make the work done a minimum, also the work done per lb. of air, and the heat carried away in the cooling water.

[Mech. Sc. Tripos, 1911.]

5. The equation for the flow of heat in the walls of an engine cylinder may be assumed to be

$$\frac{k}{c}\frac{\partial^2 V}{\partial x^2}=\frac{\partial V}{\partial t}$$

k and c being the conductivity and thermal capacity per unit volume of cast iron and V the temperature at time t at the distance x from the surface. Show that the solution of this equation is

$$V = V_1 e^{-mx}\cos(\theta - mx)$$

for a simple harmonic variation of surface temperature of semi-range V_1; θ being the angle described by the crank in time t, its angular velocity being $2\pi n$. Hence find the range at any depth x. Prove that the value of the index coefficient m is $\sqrt{(\pi nc/k)}$; and find the heat absorbed in thermal units per square foot of wall surface per period, the semi-range at the surface being V_1.

[Mech. Sc. Tripos, 1906.]

6. Two equal air receivers, charged to pressures P_1 and P_2 respectively and at the same temperature, are connected by a stop-cock. The stop-cock is opened until the pressures are equalized, and then closed. Assuming no loss of heat from the system, show that the pressure immediately after the cock is closed is $\dfrac{P_1 + P_2}{2}$.

[Mech. Sc. Tripos, 1913.]

7. State briefly the steps in the argument which justifies the statement: Of every unit of heat obtainable from a body at absolute temperature τ, at least a quantity $\dfrac{\tau_0}{\tau}$ must ultimately be wasted, τ_0 being the lowest available temperature.

A heat engine takes in 500 Th.U. at 150° C., and rejects its heat to a body whose temperature rises 1° C. for every 5 Th.U. of heat absorbed. If the temperature of the body is initially 15° C., show that the maximum work which the engine can do is 115 Th.U., and that the final temperature of the body will then be 92° C.

[Mech. Sc. Tripos, 1913.]

8. A semi-infinite solid is initially at zero temperature. Its plane face $x = 0$ is suddenly raised to temperature θ, and is maintained at that temperature. Show that after t seconds the temperature at a depth x is a function of $\dfrac{x}{\sqrt{t}}$ and find an expression for the temperature. Show further that at time t the temperature gradient at the

surface $x = o$ is $\frac{o}{\sqrt{\pi a t}}$ where $a = \frac{k}{c}$, k being the conductivity, c the specific heat, and ρ the density of the material.

[Mech. Sc. Tripos, 1912.]

9. A quantity of heat Q per unit area is suddenly communicated to the plane face of a solid slab of great thickness. Prove that the temperature t seconds later at a depth x within the solid will be

$$\alpha Q \sqrt{\frac{c}{kt}}\, e^{\frac{-cx^2}{4kt}}$$

where k is the thermal conductivity, e the thermal capacity per unit volume, and α a constant.

Plot a curve on any convenient scale showing the temperature distribution in the solid at any time, and show from this curve that $\alpha =$ 0·565 nearly.

[Mech. Sc. Tripos, 1911.]

10. Draw the entropy-temperature diagram for an ideal gas-engine working on the Otto cycle, and compare it with the similar diagram of the same cycle when the specific heats are functions of the temperatures given by $C_v = \beta + s\tau$. $C_p = \alpha + s\tau$, τ being the absolute temperature. Compare the efficiencies in the two cases.

SECTION II

GAS ENGINES AND GAS PRODUCERS

CHAPTER V

The Gas Engine

Modes of Action—Types of Engine—Humphrey Gas Pump—Gas Turbine—Methods of Improving Efficiency—Indicators and Indicator Diagrams—Heat Balance Sheets—Engine Tests—Governing—Cyclic Irregularity—Balancing.

77. Modes of Action.—The three ideal cycles of operation are :—

(1) Constant pressure cycle.
(2) Constant volume cycle.
(3) Constant temperature cycle.

And an engine working on one of these may be either a

(1) Two-stroke engine, or
(2) Four-stroke engine.

And it may further be either :—

(1) Single-acting, or
(2) Double-acting.

This leads to a considerable variety of types, although fully 80 per cent. of the internal combustion engines in the world work on the constant volume cycle, are four-stroke and single-acting. The various combinations are set out in the following table, and the names of the more important of the representatives of each class are given. Some engines, such as the Humphrey Pump, and Gas Turbines, do not fall into any distinct class.

- Internal Combustion Engines
  - Constant Pressure Cycle
    - Two-stroke
      - Single-acting (Diesel oil engine)
      - Double-acting
    - Four-stroke
      - Single-acting (Diesel oil engine)
      - Double-acting
  - Constant Volume Cycle
    - Two-stroke (Clerk cycle)
      - Single-acting (Oechelhauser gas engine and some petrol engines)
      - Double-acting (Koerting gas engine)
    - Four-stroke (Otto cycle)
      - Single-acting (most common form of engine, oil, petrol or gas. Many types)
      - Double-acting (Cockerill and several other large gas engines)
  - Constant Temp. Cycle (not used)

The relation between the maximum number of explosions per minute and the number of revolutions per minute depends upon the class to which the engine belongs. The four-stroke single-acting engine has one explosion, and therefore one working stroke for two engine revolutions; the four-stroke double-acting and the two-stroke single-acting engines have one explosion per revolution, and the two-stroke double-acting engines two working strokes per revolution, as in the case of a steam engine.

Practically all gas engines work on the constant volume cycle, sometimes two-stroke but more often four-stroke; the former is commonly known as the Clerk cycle and the latter the Otto cycle, from the names of those who first suggested them.

78. Typical Otto Cycle Engine.—A typical instance of the well-known Otto cycle gas engine is shown in section in Fig. 29. Gas and air enter through the casting G, and pass the inlet valve A (operated by cam arrangements not shown in the figure). As the gas and air enter, the piston C moves along the cylinder, being drawn by the connecting rod D attached to the crank-shaft E. After the gases have been

allowed to fill the cylinder they are compressed on the return stroke of the piston, are ignited on the inner dead-centre,

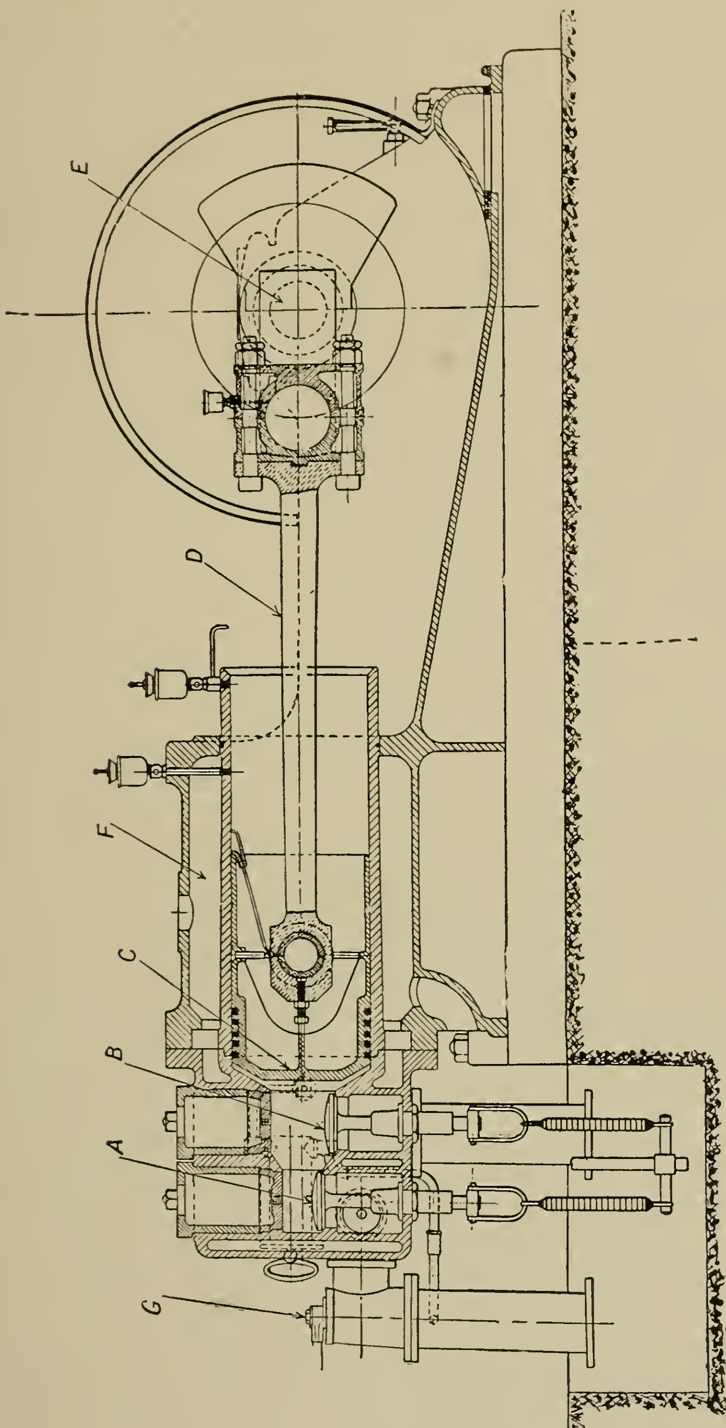


Fig. 29.—The Grice Gas Engine.

expand doing work, and on the exhaust stroke the exhaust valve B opens to allow them to escape into the silencer. The cylinder is kept sufficiently cool, to enable it to be lubricated, by means of a circulation of water in the jacket space F.

79. The Origin of the Clerk Cycle.—This cycle was invented some years after the Otto by Dugald Clerk, who gave the following description of it to the Society of Arts in 1905: "In the Clerk engine the motor cylinder had, at the front ends, large ports leading into an annular space, these being the exhaust ports. The compression space was conical, and the charge was sent in by means of a separate pump, which I called the displacer. The action of the engine was as follows: When the piston got to the out end of its stroke, and the crank was crossing the out centre, the piston overran the exhaust ports on the out-stroke, and covered them on the in-stroke. Meantime the pump or the displacer piston, which was attached to a crank at right angles in advance of the main crank, was sweeping in and giving its charge a slight compression. That charge passed through a connecting pipe, and through a check valve, into the conical end, displacing before it all the contents of the cylinder. When the main crank had returned about 40 degrees of its circle under the centre, these ports were closed. It opened about 40 degrees above and closed 40 degrees under, and in that time the displacer piston had gone fully in and discharged its charge into the cylinder and combustion space through the lift valve. Then the motor piston compressed the charge, and ignition took place at the in-end of the stroke, just as in the Otto cycle. The object of the invention was to enable one motor cylinder to give an impulse at every revolution. In the Otto cycle there is only one impulse for two revolutions, so far as the main cylinder is concerned. The Clerk engine gave one impulse for every revolution of the main crank in the main cylinder, but to make that possible it was necessary to provide an auxiliary crank and displacer cylinder.* The idea was, of course, to diminish the irregularity of the Otto cycle by having

* Crankcase compression is now generally adopted for small two-stroke engines, thereby avoiding the necessity for a separate displacer cylinder.

an impulse at every revolution, or more frequently, that is to say, two impulses per revolution, obtained by making the engine double-acting. The object was to get very much more power for a given weight of engine, as the pump was light and only required to deal with its charge at a low pressure."

80. Large two-stroke Engines.—There are two well-known

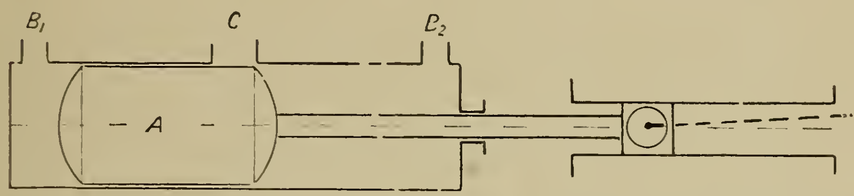


FIG. 30.—Diagram of Koerting Engine. B_1 and B_2 Inlet Valves ; C Exhaust Valve.

designs of large two-stroke engines, both German in origin, viz., the Koerting and the Oechelhauser. The former is double-acting and the latter single-acting. In the Koerting engine, illustrated digrammatically in Fig. 30, the piston A is unusually long, and by alternately covering and uncovering the exhaust port C is itself a part of the valve gear. The mixture of gas and air is admitted alternately at B_1 and B_2 from the pump cylinder. If there has just been an explosion at the left-hand end of the cylinder, the piston will move to the right, and the gases will expand, doing work. By the time the piston has got to the end of the stroke, it will have uncovered the exhaust port C, and the burnt gases will rush away. At this point the inlet valve B_1 is opened, and fresh air and gas are pumped in. This fresh charge is compressed on the return stroke of the piston, and ignited, as before, on the dead point. The same cycle of events goes on at the right-hand end of the cylinder, so that there is an explosion at the right-hand end when C is uncovered in the left-hand end and vice versa. The piston receives therefore just as many impulses as a steam engine piston.

An obvious difficulty about this method of working is that some of the incoming gas may be caught up by, and pass away with, the exhaust products and be lost. This reduces economy, but is of little importance when working with what

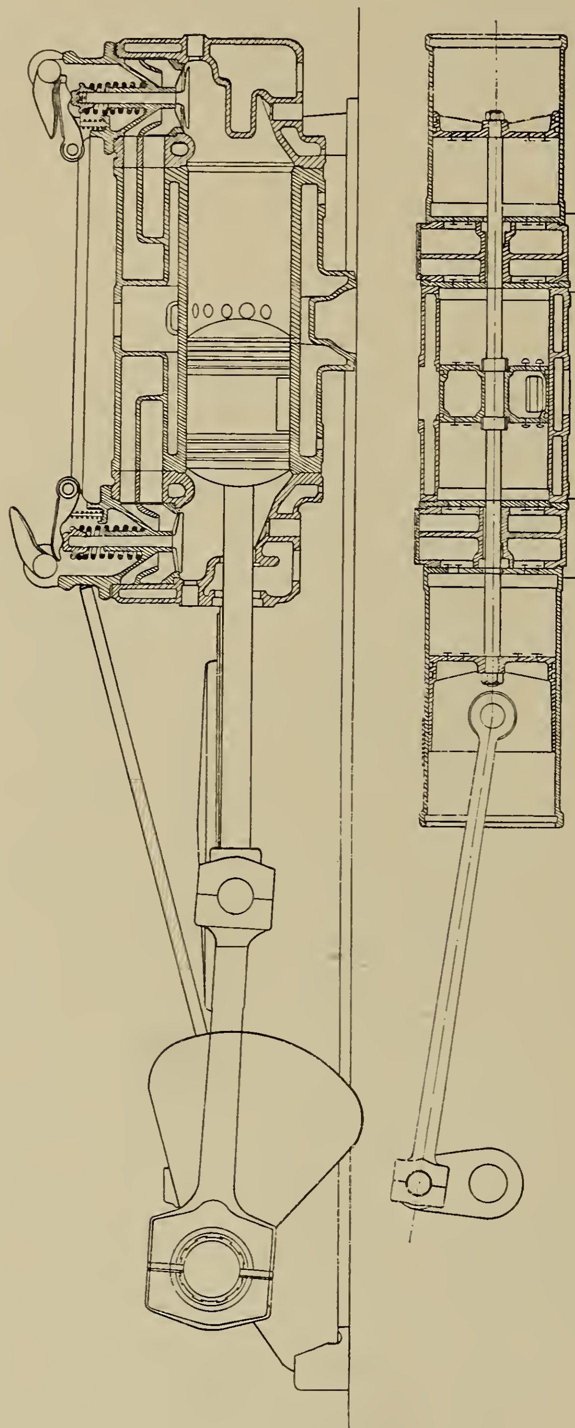


FIG. 31.—Longitudinal Section of Two-Cycle Engine (Koerting)—by Mather and Platt. Power cylinder shown above. Gas and Air pump below. Note inlet valves at each end and length of piston in power cylinder.

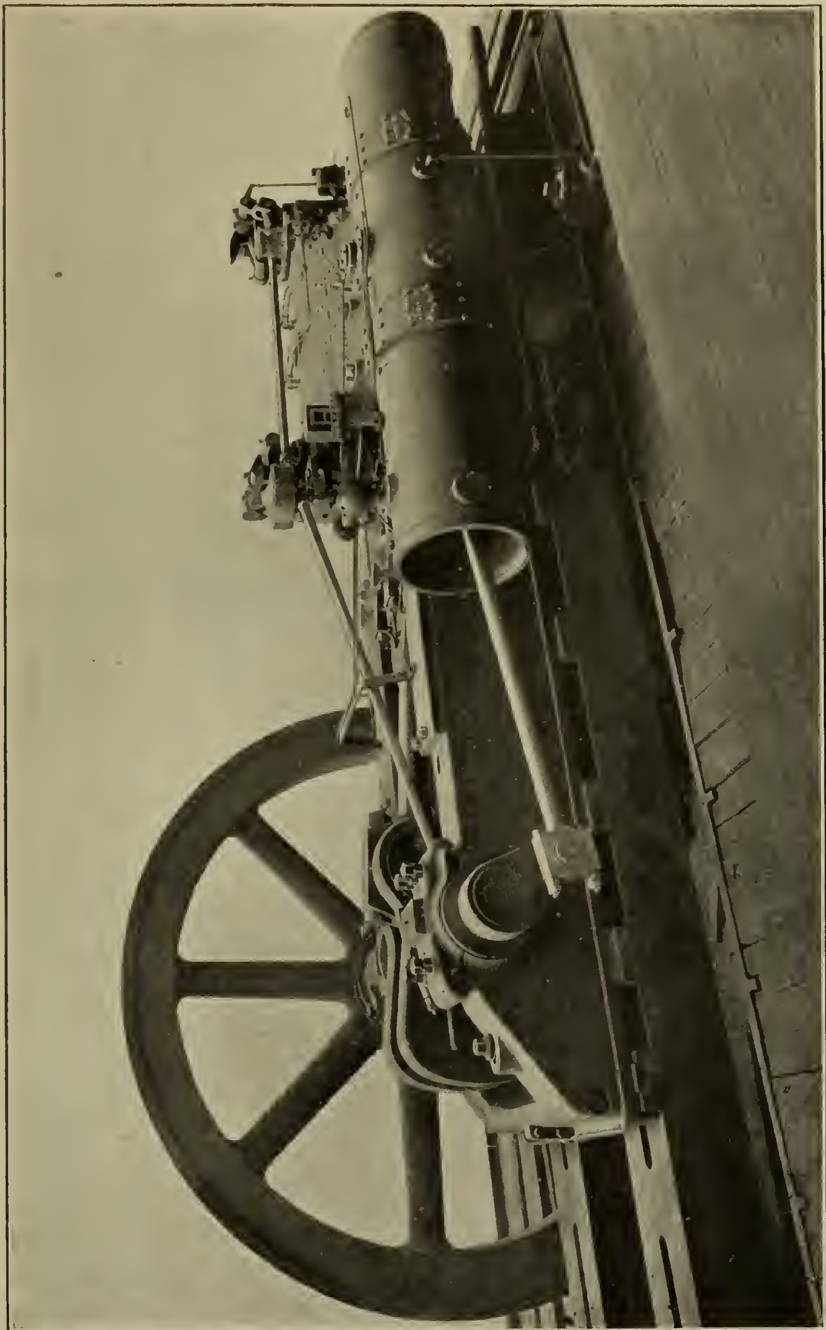


FIG. 32.—300 B.H.P. Koerting Gas Engine (Mather and Platt). Note inlet valves operated by eccentric on main shaft.

are known as *waste gases,** as these are produced in immense volume and at practically no cost.

* *See* Ch. VII.

Attention has been given in the previous chapter to the question of piston and cylinder wall temperatures, and it will therefore be readily understood that in such a cycle as this the heating effect of explosions so closely following each other will be severely felt and high temperatures are likely to be reached by all parts open to the gases.

In the Oechelhauser engine shown in Fig. 33, there are two pistons which are so connected to the crank-shaft as to move alternately towards and away from one another. Air is admitted from the pump cylinder through the holes C, in the

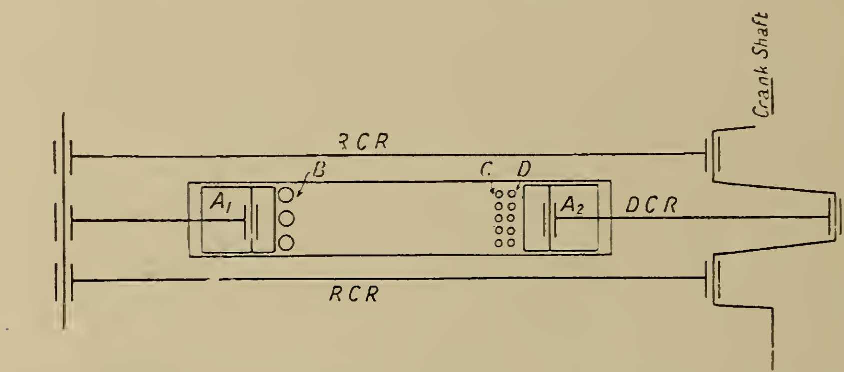


FIG. 33.—Diagram of Oechelhauser Engine. *D C R* direct Connecting Rod; *R C R* two Return Connecting Rods; Air enters by Holes at *C*; Gas at *D*; Exhaust leaves at *B*.

walls of the cylinder, and gas through the holes at D as the piston, A_2, uncovers them in turn. The mixture is then compressed between the two pistons as they come together, ignited at the dead point, expanded during the expansion stroke, and then exhausted by the uncovering of the exhaust port B. The engine is single-acting so that only half the horse-power is obtainable as compared with a Koerting engine of the same cylinder dimensions. The piston A_2 works directly on to the crank-shaft as shown, but the piston A_1 has to have a separate cross-head and two long return connecting rods. The balancing is good, as the two heavy pistons balance one another. The Gobron-Brillée and an early form of the Arrol-Johnson motor car engines worked in much the same way. The engine invented by H. F. Fullagar * and

* H. F. Fullagar, *Proc. I.M.E.*, 1914.

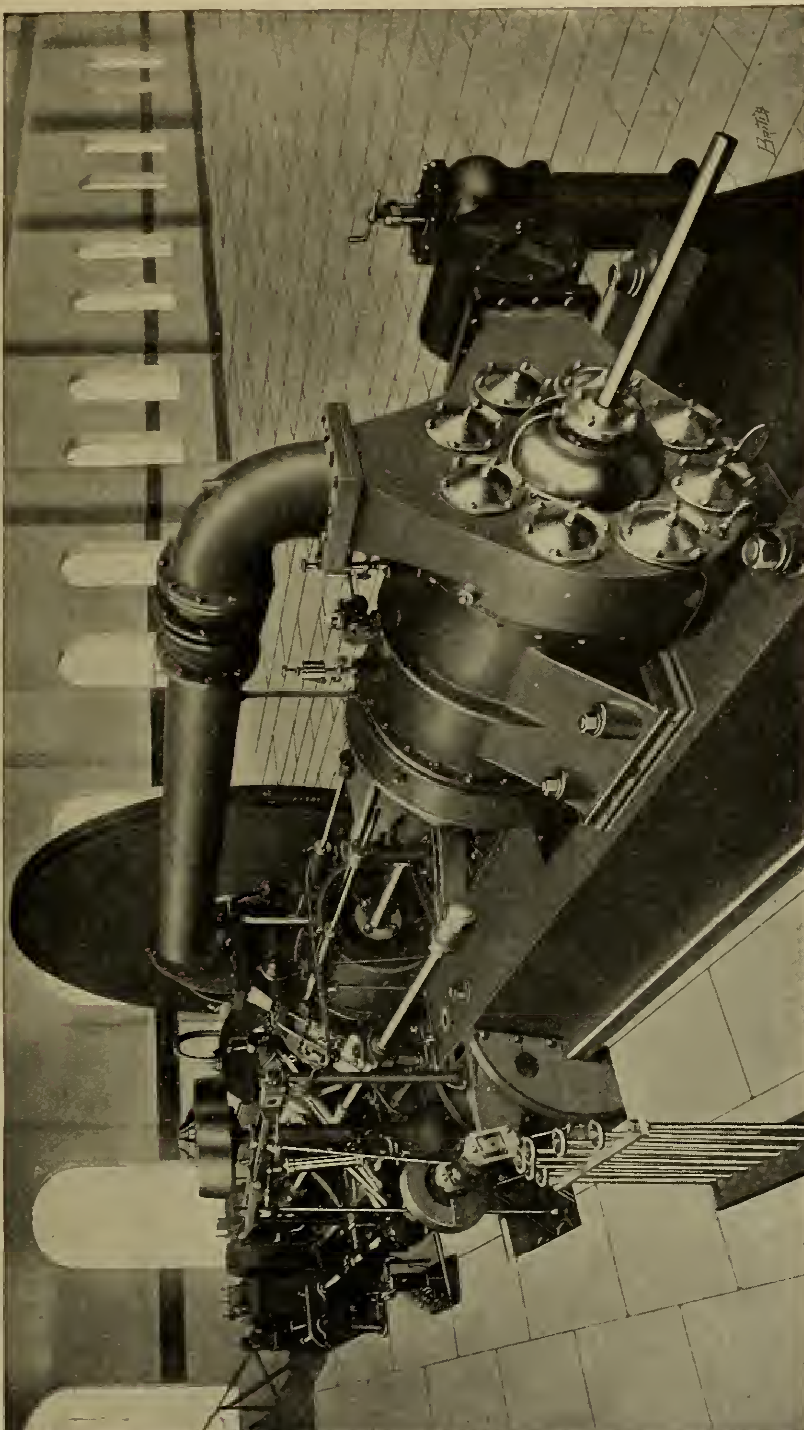


FIG. 34.—500 H.P. Oechelhauser Engine coupled to Dynamo, giving 440 volts and 850 ampères at 105 r.p.m.

known by his name also bears a general resemblance to the Oechelhauser type.

That there are difficulties to be overcome in the construction and working of large gas engines, particularly those working on the two-stroke cycle, is shown by the history of the generating station for lighting and tramway work which was established some years ago at Johannesburg. The plant was operated with Oechelhauser gas engines driven by gas from Poetter producers using Transvaal coal. From the beginning many difficulties arose in the operation of this plant, mainly on account of the lack of requisite experience in the manufacture and operation of similar installations. The cost per unit is reported to have been higher than was anticipated, and the plant has since been shut down. This occurrence must not be taken to show that Oechelhauser engines are unsuitable for heavy work, since in numerous places on the Continent engines of this make are working well.

A large installation of gas engine plant in this country is that at the Cargo Fleet Iron Co.'s works at Middlesbrough. It consists of six **Cockerill** engines, built by Messrs. Richardsons, Westgarth & Co., of 900 H.P. each, or a total of 5,400 H.P. These engines work on the Otto cycle, but by having double-acting tandem cylinders the crank gets just as many impulses as in a double-acting steam engine.

81. Other Types of Engines.—There are a number of British and foreign makes of engine (e.g. the Ehrhardt and Sehmer engine) which work on the four-cycle double-acting principle, giving an explosion per revolution for each cylinder, so that for two cylinders placed tandem every stroke is a working stroke just as in a steam engine. With the tandem arrangement there is the advantage that the motion towards each dead centre is always accompanied by a compression stroke; this leads to a cushioning action which is useful as an aid to overcoming the inertia effects due to the moving parts. Engines of this type have met with much success, thus Mathot,* writing in *Gas and Oil Power*, stated that: "A 600 H.P. double-acting Ehrhardt and Sehmer engine, with two tandem cylinders, was tested, without previous cleaning, after four months'

* December 15, 1906.

continuous work with coke-oven gas of from 4,000 to 4,200 calories; the trial was carried out by the makers' engineers under the supervision of Kgl. Berginspection's engineer. Thanks to a large gasometer, the record of gas consumption was taken for a period of one hour. The mechanical efficiency was 83 per cent. The engine was new and was tested on a normal load. The dimensions of its pistons and piston rods were respectively 620 mm. and 170 mm., 750 mm. stroke, and 150 revolutions. It just reached 520 KW., generating three-phase current. In the conditions of the trial the actual thermal efficiency was more than 31 per cent., or nearly $37\frac{1}{2}$ per cent. of that indicated. Unfortunately, similar trials to this are rare because gasometers are not usually sufficiently large to measure the exact amount of gas used by the engine."

Another well-known type is the **Premier gas engine.** Perhaps its most familiar feature is the scavenging of the exhaust products by means of an air blast. It has been stated that this blast is capable of keeping the cylinder interior almost free from deposit even when working with bituminous fuel gas plant. An account has been published * of a 1,200 H.P. four-cylinder gas engine by the Premier Gas Engine Co., which was constructed for direct coupling to a continuous current generator and consisted of two sets of tandem cylinders working on cranks set at 180 degrees apart. A four-stroke cycle was employed, so there were two working strokes per revolution. A scavenging charge of air supplied from a separate air cylinder at about 3 lb. per square inch was used to clear the cylinders of waste products. All the valves were placed on the cylinder covers; pistons and exhaust valves were both water-cooled. It was stated that, operating on producer gas, a compression pressure of 140 lb. per square inch could be used without any difficulty whatever from pre-ignition, and that a test on the engine showed the mechanical efficiency to be as much as 87 per cent.

82. The Humphrey Gas Pump.—One of the most remarkable applications of the internal combustion principle is seen in the gas pump invented by H. A. Humphrey, and described by him at a meeting of the Institution of Mechanical Engineers

* *Engineering*, Jan. 11, 1907.

in 1909. Its chief feature is the use of a swinging column of water in place of piston and connecting rod, the upper face of the water column taking the place of the piston face.

The following extract from the inventor's description * makes his purpose clear :—

"The idea of exploding a combustible mixture of gas and air to produce pressure on the surface of water, with the object of raising the water, is of course not new, and attempts to put this idea into practice date back to 1868. The efforts have all been directed too much along the lines of ordinary pumps, in so far that the water lifted has always been forced past a non-return valve, and the operation of such a valve with the explosive force behind it has been inevitably disastrous. In the types of pumps invented by the author there is, when the explosion occurs, a full-bore passage from the combustion chamber to the final outlet, also some of the water pumped to a high level by the energy of the explosion is allowed to return again to compress a fresh combustible charge. When sudden changes of velocity occur in masses of a heavy and incompressible liquid, like water, great difficulty is found in controlling the movement of the liquid. All such difficulties are removed in the author's pumps by allowing the movements of liquid to control the pump, and by causing the mass of liquid moved to be sufficiently large, so that the velocities are never excessive. The mass of water forms a pendulum which swings between the high and low level, and, by its movement alone, serves to draw in fresh water, to exhaust the burnt products, to draw in a fresh combustible charge, and to compress the charge previous to ignition. With the movements of the liquid quite unrestrained by any of the usual mechanical appliances, the result is a pump which works with freedom from shock and noise, and which requires very few working parts.

"The subject attains a wider scientific interest from the fact that the apparatus follows a cycle in which the expansion of the burnt products is carried to atmospheric pressure, and so involves a thermodynamic cycle of greater efficiency than can be claimed for the Otto cycle."

It is stated that the pump is suitable for working with

* *Proc. I.M.E.*, 1909.

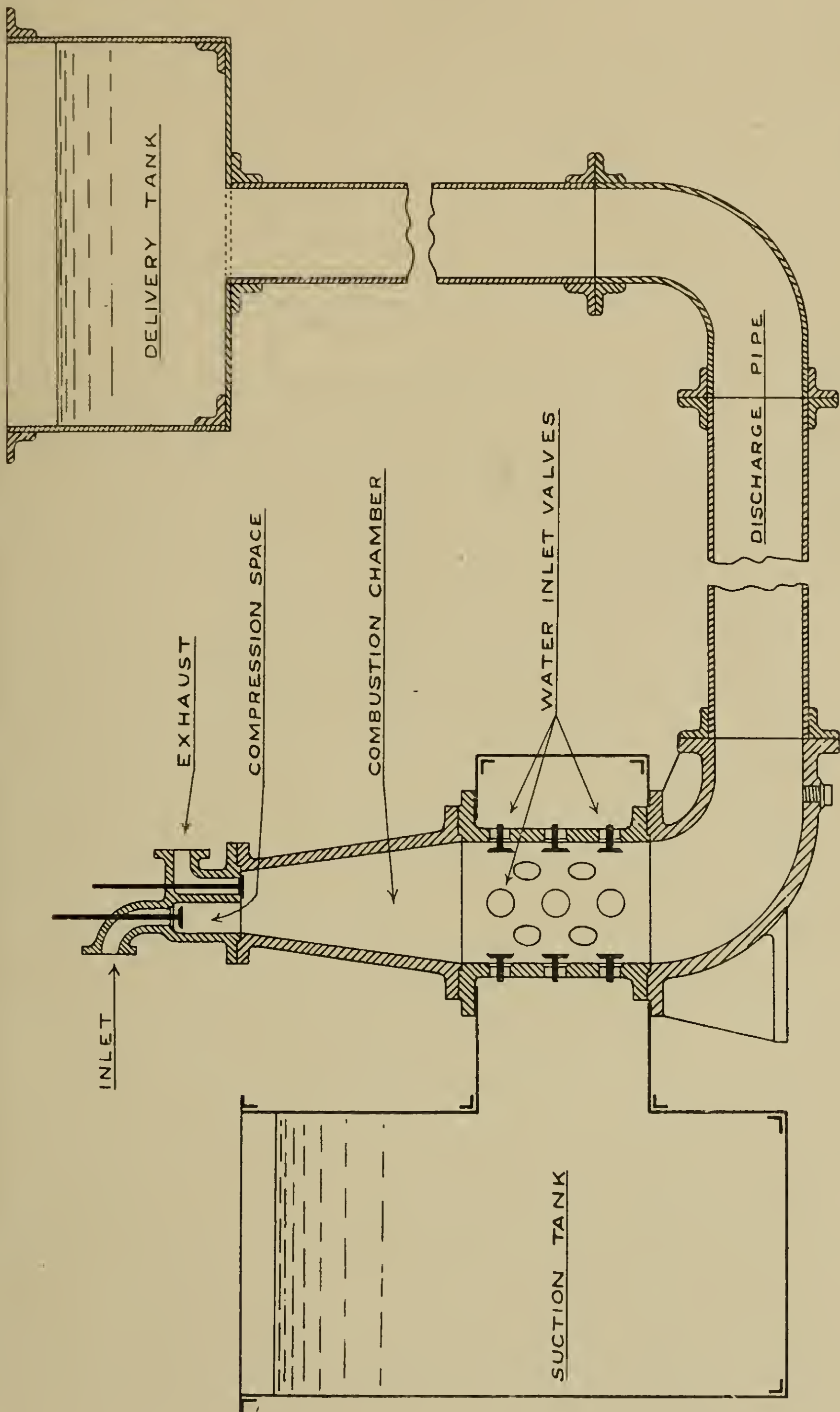


FIG. 35.—The Humphrey Pump.

producer gas, suction gas, lighting gas, petrol or paraffin; there are no moving parts except the interlocked inlet and

exhaust valves to the combustion chamber and the outlet valves from the suction water-tank. The use of a flywheel is not necessary, since the reciprocating water column performs also the flywheel function.

The cycle followed is a constant volume cycle, but the strokes are not all of equal length. The strokes are, (1) a long stroke during explosion and expansion, (2) another long stroke during exhaust, (3) a short stroke during suction, and (4) a still shorter stroke during compression.

A simple form of this pump is shown diagrammatically in Fig. 35.

Imagine a charge of gas and air to be compressed at the top of the combustion chamber in the space shown and let it be fired by a sparking plug, not shown in the diagram. The explosion pressure drives the water out of the combustion chamber and sets the whole water column in the discharge pipe in motion. As the gases expand the kinetic energy of the moving water steadily rises, until the gases arrive at atmospheric pressure. There is then no further driving force on the water column, but owing to its kinetic energy it continues to move forward and the pressure in the combustion chamber sinks below the atmospheric pressure, so opening the exhaust valve and the water outlet valves from the suction tank. Water rushes from this tank into the discharge pipe and into the combustion chamber. When the kinetic energy of the moving column has entirely expended itself in forcing water into the delivery tank and in overcoming the friction due to its motion, it comes to rest and gradually starts to swing back again until it rises to the upper part of the combustion chamber and closes the exhaust valve by impact. Its further motion is arrested by the cushioning effect of the burnt products trapped in the compression space, and gradually the water-pendulum begins to swing back again, opening the inlet valve and drawing in a fresh explosive charge which on the return swing of the water is compressed, and, at the dead-point, fired, so bringing a fresh cycle of events into operation. Thus the pumping continues.

A test by Unwin in 1909 showed a consumption of 1·063 lb. of anthracite in the producer per pump H.P. hour correspond-

ing to 12,243 B.Th.U. per P.H.P. hour, a very satisfactory result. A large battery of these pumps was subsequently erected at the Chingford Reservoir * under a guarantee that not more than 1·1 lb. of anthracite per P.H.P. hour would be consumed. Later still a very large plant has been constructed for Egypt †; eighteen pumps will be needed, each capable of delivering 100,000,000 gallons per day through a lift of 20 ft. The combustion chambers are 8 ft. 8 in. in diameter and about 14 ft. high.

83. The Gas Turbine.—The great success achieved by the steam turbine has naturally led inventors to make numerous attempts to devise a gas turbine, and so to reap the joint advantage of the high thermal efficiency of the gas engine and the even rotary motion of the steam turbine. A gas turbine may be defined as a turbine in which a gas, usually a product of combustion, with or without an admixture of steam, is the working medium. This medium suffers, however, from the disadvantage that it is not condensible as is steam, and it is not therefore possible to obtain the "toe of the diagram" unless the exhaust products are drawn out by a pump (in which event care has to be taken that the gain due to increased expansion is not overbalanced by the work of pumping).

Gas turbines are divided into two main types, those in which combustion occurs at constant pressure and those in which it occurs at constant volume; the latter are known as explosion turbines. Hybrid machines operate on an intermediate cycle and share the features peculiar to both.

Much of the difficulty in all types of turbine is the burning of the blades owing to the high temperature of the gases. This temperature is reducible by the addition of steam, but unless this steam is afterwards condensed the latent heat in it is lost.

The number of gas turbines hitherto built is very small. The first, due to Réné Armengaud, was begun about 1904, and gave as much as 300 B.H.P., but the efficiency was very low. It operated on the constant pressure method. The

* *Engineering,* February 14, 1913.
† *Internal Combustion Engineering,* June 24, 1914.

next in order of time was the little explosion turbine, giving less than 2 H.P., designed by Karovodine; despite its small size it promised well for small unit work. The third was the 1,000 H.P. explosion turbine of Holzwarth's which met with a fair amount of success, although it did not yield more than half of the 1,000 H.P. it was designed to give.

Dugald Clerk * thinks there is little chance for the constant pressure turbine, but that better prospects lie before the type of turbine in which successive gas explosions propel jets of water against the vanes of a water turbine and in which there is therefore no contact of flame and blade. Norman Davey † considers on the other hand that blade temperature, even when in direct contact with flame, can be kept within manageable limits, so offering prospect of success for a constant pressure mixed (steam and gas) fluid working at very low pressures.

Modern steam turbines work with steam which is highly superheated and which is therefore in the state of a gas and not a vapour. Such turbines may therefore be called gas turbines, although not on the internal combustion principle. Such "external combustion gas turbines," if so they may be called, may prove too successful for the competition of the internal combustion turbine, especially if, owing to the great improvements being effected in steam boilers, the efficiency of the latter can be brought up to the level reached by the gas-producer. This question formed a part of the subject of Ferranti's "James Watt Lecture," ‡ and the following extract is quoted:—

"The speaker began experimenting some years ago, and had now, after many failures and the expenditure of much money and time, produced a turbine which at the highest temperatures and with great and rapid variations of temperature was quite free from mechanical troubles.

"In this turbine the steam was superheated initially, and after the first expansion and while it was still superheated it was re-superheated before it did its work in the second stage

* British Association, 1912.
† *The Gas Turbine*, 1914.
‡ *Times*, January 22, 1913.

of the turbine. After this it was exhausted in a superheated condition through a regenerator to the condenser. The whole of the blading was electrically welded to avoid the straining due to caulking at the high temperatures that were reached and also the loosening that occurred from the same cause. The blading was formed of mild steel with a thin coating of pure sheet nickel electrically welded on the surface, was most accurately finished to shape by a process of step-by-step pressing under very heavy pressure, and was welded accurately in position.

"The steam was worked as a gas at a high temperature throughout the turbine, and this coupled with the many improvements above referred to had given very good results. The 5,000-H.P. machine, which had now been running for some time, when tested at a load of two-thirds full power had given a shaft H.P. on 7 lb. of steam which, if supplied by an oil-fired boiler superheater system of 85 per cent. efficiency, which had already been exceeded in central station practice, would consume less than ·625 lb. of oil per S.H.P. From many tests already made it appeared that when it was run at full load under favourable conditions it would take less than 6 lb. of steam per S.H.P., and that the system under the conditions named would have a thermal efficiency of over 24 per cent., corresponding to an oil consumption of about ·55 lb. of oil per S.H.P. The tests were being continued, but as the turbine was supplying power continuously to a large works with a constantly varying load, it was not easy to do what was necessary to enable tests to be carried out. So far as he could see, the system, when applied on a large scale, would be capable of giving an over-all thermal efficiency of 29 per cent."

84. Methods of Improving Efficiency (Crossley and National).—One of the chief causes which limit the efficiency of gas engines is the high temperature during explosion and the very rapid rate at which heat is then abstracted by the walls. Two methods have been tried with a view to minimize this effect, the idea in each case being to reduce the maximum temperature of the cycle without, however, decreasing the mean pressure. These two are the water

injection method of Messrs. Crossley Bros., and the super-compression method of Dugald Clerk.

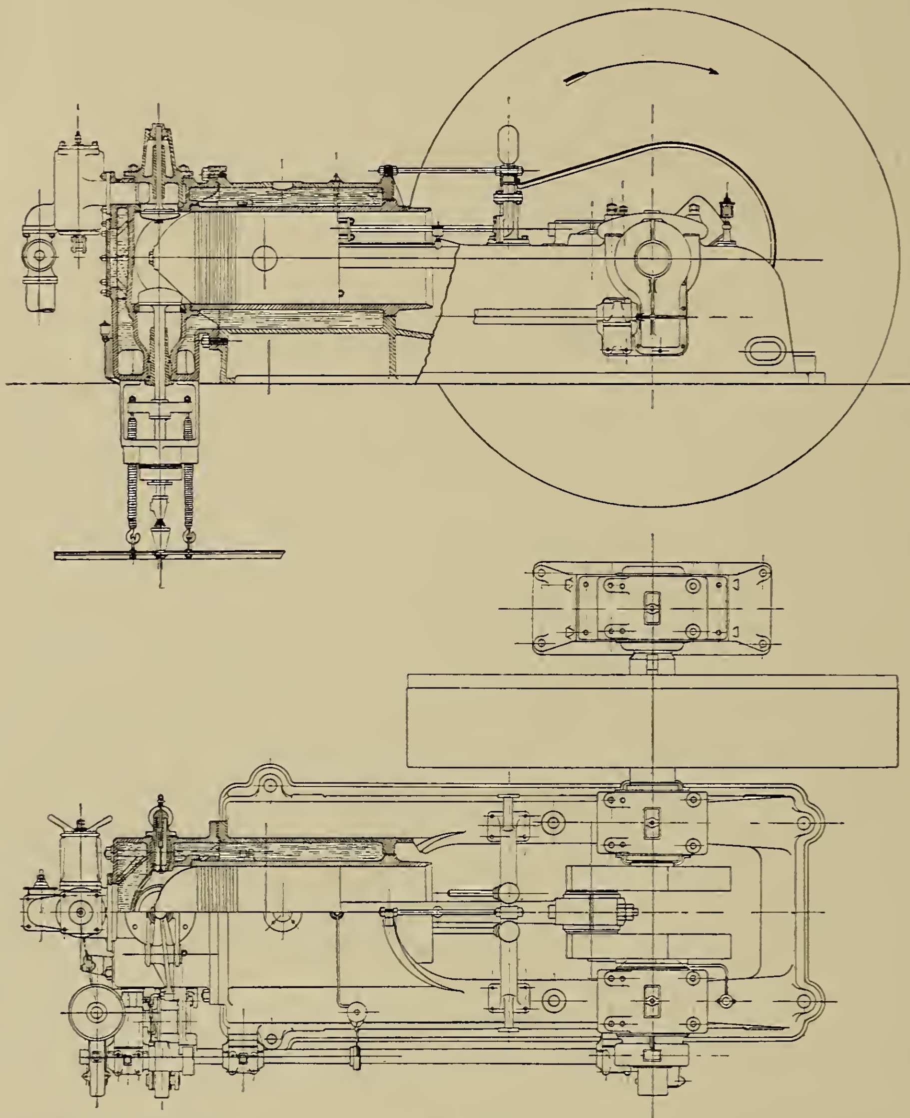


FIG. 36.—Campbell Gas Engine. Note the disposition of the inlet and exhaust valves, and the water cooling arrangements. On the plan at the lower side is seen the half-time-shaft.

85. The Water Injection Method.—Messrs. Crossley decided to try the effect of injecting a small spray of water into the cylinder during the suction stroke. The water, entering as a fine spray in part of the air supply, was as evenly distributed as possible and did not form a water film on the cylinder walls.

Very little water is required, because of the high value of its latent heat. As the mixture explodes the water mist is evaporated into steam and the heat so absorbed prevents the temperature of the gases from rising unduly high. A 50 H.P. engine so adapted was tested by Professor Burstall in 1904 and the following records taken :—

Size of engine	14 in. × 21 in.
Average revs./min. . . .	166·02
,, I.H.P.	60·5
,, B.H.P.	49·7
Mechanical efficiency . . .	82·2 per cent.
Gas used per I.H.P.-hour . .	11·77 cu. ft.
,, ,, B.H.P.-hour . .	14·43 ,,
Calorific value of gas (lower value)	578 B.Th.U. per cu. ft.
Thermal efficiency on I.H.P. .	37·43 per cent.
,, ,, ,, B.H.P. .	30·8 ,,
Water used in cylinder . .	0·131 lb. per minute.
,, discharged from jacket .	25·66 ,, ,,
Mean temperature of exhaust as measured by Callendar pyrometer	718° F.

The ratio of air to gas was 10·2 and the compression ratio 8·7, corresponding on the "air standard" to an efficiency of 0·58. As the actual efficiency found was 0·37 it follows that the engine achieved nearly 64 per cent. of the "air standard" efficiency. This is a higher ratio than any of those given by Dugald Clerk in his 1907 paper before the Institution of Civil Engineers ("On the Limits of Thermal Efficiency in Internal Combustion Motors"), which showed no higher ratio than 59 per cent. and that only in the case of a maximum temperature of 1,098° C., whereas when the temperature rose to 1,750° the ratio fell to 50 per cent. and below. On this method of comparison, therefore, the water injection method shows to advantage.

Water injection has been tried on a still more extensive scale by B. Hopkinson,* who used it to replace entirely the usual water jacket system. The work of the Gaseous Explo-

* "A new method of cooling gas engines," *Proc. I.M.E.*, 1913.

sions Committee had shown that almost the whole of the cooling loss occurred during and immediately after the explosion, and that the cooling of that part of the cylinder walls which did not form part of the combustion space was chiefly useful in keeping the piston cool (since unless the circumference of the piston were kept cool the centre might rise in temperature to the pre-ignition point), so that if a jet of water could be projected on to the hot face of the piston and on to the walls of the combustion space it might be possible to dispense with any other cooling arrangements altogether. This makes for great simplicity in construction and corresponding economy in first cost.

In Hopkinson's engine cold water is injected through a hollow casting projecting into the combustion chamber and provided with a number of holes or small nozzles rather less than a millimetre in diameter. The jets thus formed are comparatively coarse, and even after passing through the flaming gas most of the liquid reaches the hot metal surface upon which it is directed. These jets are directed against all parts of the surface of the combustion chamber and against the face of the piston.

The engine used for this experiment was an $11\frac{1}{2}$ in. × 21 in. Crossley engine having a compression ratio of 6·37. The following is an extract from the report on the trial :—

"The engine was run continuously for 120 hours on an electrical load with coal-gas. The engine developed during this period 43 B.H.P. on the average, and ran very smoothly and steadily. The average mean effective pressure was 101 lb. per sq. in. When jacketed, the engine would not develop more than 40 B.H.P. continuously without overheating, and mixtures giving a mean pressure of more than 100 lb. per sq. in. produced excessive maximum pressures (over 500 lb.) with violent thumping explosions. The reduction in maximum pressure, under these circumstances, by water-injection is over 100 lb. per sq. in., and the effect is very marked, the explosion becoming almost inaudible. This effect of the presence of steam in the explosive charge is, of course, well known, but the quantity of steam formed in an engine cooled in this manner is so large that it constitutes a

substantial advantage of the method. It will be noticed that the formation of the steam does not involve any thermodynamic loss, such as occurs when water is sprayed into the cylinder in an atomised condition and evaporated before reaching the walls, since the heat used is that which would otherwise be wasted in the jacket-water. The quantity of water used on this trial was, on the average, 102 lb. per hour, equivalent to 2·4 lb. per B.H.P.-hour. The temperature of the engine varied from 150° to 180° C. No water was visible on the piston or the spindles of the valves, and when the engine was stopped at the end of the trial the inside of the combustion-chamber was found to be perfectly dry. When the engine was jacketed, and giving the same power for short periods, the jacket-water removed about 67,000 B.Th.U. per hour, which would be sufficient to evaporate 108 lb. of water at a temperature of 20° C. under atmospheric pressure. The agreement between the available heat and the amount of water evaporated is satisfactory, such difference as there is being accounted for partly by greater radiation loss, consequent on the higher temperature of the engine, and partly by the reduction in flame temperature produced by the steam, which somewhat reduces the total amount of heat passing into the walls.

"The engine consumed in this trial 15 cub. ft. of Cambridge coal-gas per B.H.P. hour, reckoned at atmospheric temperature and pressure. This is approximately the same as it burnt when developing the same power for short periods when jacketed. Tests at other loads have shown that with a weak mixture the gas consumption is slightly increased by the water injection, but with very strong mixtures it is a trifle less. The difference, however, does not exceed 5 per cent. either way, and on the average it may be said that the economy is unaffected by the use of this method of cooling. Indicator diagrams taken in this long trial compared with similar diagrams taken from the jacketed engine shows that the reduction in maximum pressure is counterbalanced by a slightly raised expansion line. The pressure is better sustained, partly by the formation of the steam and partly by the reduced loss of heat, with the result that the diagram is 'fatter' and less 'peaky.'"

The method has since been successfully applied to much larger engines.

86. The Super-Compression Method.—This method is due to Dugald Clerk, who in his James Forrest Lecture (1904) before the Institution of Civil Engineers, described it thus: "Some time ago it appeared to me possible to reduce maximum temperatures by increasing the charge weight per stroke given to an engine. I had experimented with two engines, one having a 7 in. cylinder, 15 in. stroke, and the other a 10 in. cylinder, 18 in. stroke. These engines, which are of the ordinary standard four-cycle type, are allowed to take in the usual charge of gas and air; then at the end of the stroke a further charge of air or other inert fluid is added to increase the pressure in the cylinder to 7 lb. or 8 lb. per square inch above atmosphere before the return of the piston. A small part of the return stroke is, however, made before the pressure can be materially increased as the added charge takes some time to fill the cylinder. This has the effect of increasing the charge weight present in the cylinder by about 40 per cent. and of increasing the pressure of compression without, however, increasing the temperature of compression. Indeed in both experiments the temperature of compression was diminished. As the charge present is constant so far as gas is concerned, the maximum temperature capable of being produced is much reduced. The maximum temperature shown by the diagrams taken by me from these two engines is about 1,200° C. Experiments were made and it was found that the heat-flow was reduced to about two-thirds, and further that the mean available pressure was increased about 20 per cent."

The thermal efficiency of an engine which on working without super-compression was 27·7 per cent. showed an increase to 34·4 per cent. when super-compression was adopted. One sees therefore that if the atmospheric pressure were 50 per cent. higher than it is, it would suit the working of gas engines a great deal better.*

* The converse of this is seen in the case of engines which have to work at stations which are at a considerable height above sea level. The horse-power under these conditions is stated to fall off by 3 per cent. for every 1,000 ft. of altitude, and this is the usual allowance made by manufacturers.

The improvements in operation obtained by the water injection and the super-compression methods are of course desirable in themselves, but they are really the most welcome for what they bring in their train, viz. freedom from cracking of cylinders and pistons. Whenever a large amount of heat has to be passed through the walls to the cooling water (and the larger the engine the larger the amount of heat to be got rid of in this way and the smaller in proportion to cubic contents is the cooling surface), there arises a steep heat gradient in the metal in contact with the gas, which in turn leads to differential expansion and the consequent failure of engines owing to the cracking of ends or walls or sometimes of pistons themselves. Manufacturers who seek high thermal efficiencies seek them not so much for the resulting economy in fuel, but for the increased freedom from mechanical difficulties in operation. Improved methods which allow of the maximum cyclic temperature being reduced without any loss of power can also be pressed in the direction of increasing the mean pressure considerably without, however, raising the temperature so high as it was previously. This leads to greater output, but the pressure at exhaust is considerable, and it would in such cases be an advantage to use this exhaust in another cylinder and so compound the engine. Efforts in this direction have been made, but the difficulties of construction are great.

87. The Indicator.—A very important instrument used in connexion with gas engines is the indicator. It is an apparatus which when attached to an engine draws a curve showing how the pressure in the cylinder varies at different points in the stroke. The best known modern form is the Crosby shown in Fig. 37. On the left of the illustration will be seen a small cylinder containing a cup-shaped piston which is regulated in its upward motion by the downward push of the strong spring seen above. When the indicator is screwed on to the engine cylinder the gas pressure causes the indicator piston to rise through a distance proportional to the force exerted. The little piston rod rises also and communicates its motion to the long sloping lever seen above. This lever carries at its far end a pencil which traces a line on a paper

sheet fastened round the drum seen on the right which is made to oscillate to and fro by the cord shown on the extreme right of the diagram being attached to a moving link which has a motion similar to that of the piston, but less in amount. The pencil therefore traces out the closed curve known as indicator diagram.

Fig. 38 shows another form of the instrument having the *spring outside*, where it is less affected by the heat and so gives a better reading. Of course all these springs need to be care-

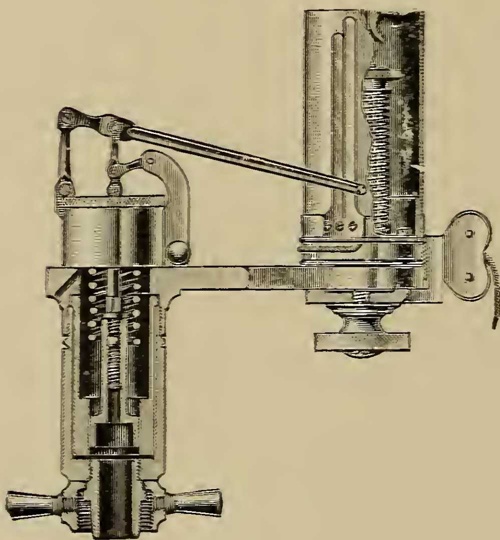


FIG. 37.—Crosby Indicator with Internal Spring.

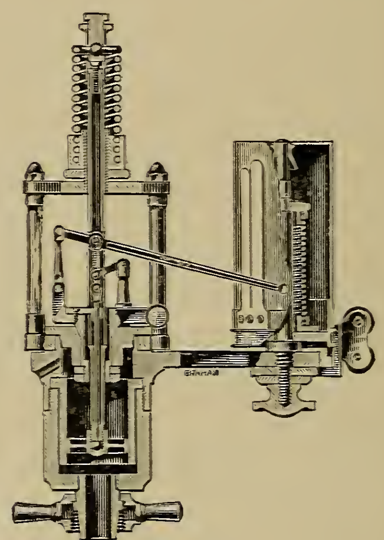


FIG. 38.—Crosby Indicator with External Spring.

fully calibrated in advance so that it is known how much pressure is represented by a rise of the pencil point equal to, say, 1 inch. There are certain qualities which a well-designed indicator should have. It must have a spring stiff enough to ensure that the maximum pressure will come well within its range. It must have a well-designed piston, as light as is consistent with strength, which will move freely in the cylinder. A slight leakage of gas is much less of an evil than any chance of the piston sticking or jambing. In the Crosby form the piston is made from a solid piece of tool steel, hardened and then ground, and lapped to gauge. It is provided with a socket to receive the bead at the end of the spring and has

screw adjustments for locking the spring in place. As has already been indicated, the to and fro motion of the paper is obtained from a cord attached at its other end to some point in the upper part of a swinging lever of which the lowest point is connected with the engine piston or some part that moves with it, so that the motion of the engine piston is reproduced to a convenient scale. There is also a Crosby reducing device for doing this. It is illustrated in Fig. 39, and its principle of action is easily seen therefrom. In this device the cord at the bottom can be fastened direct to the crosshead, or other part attached to the piston, the cord passing over guide pulleys if necessary. It is better, however, not to have a longer cord than necessary, lest its stretching with the pull put on it should introduce error in the indicator card. For very accurate work the cord is sometimes replaced by steel wire.

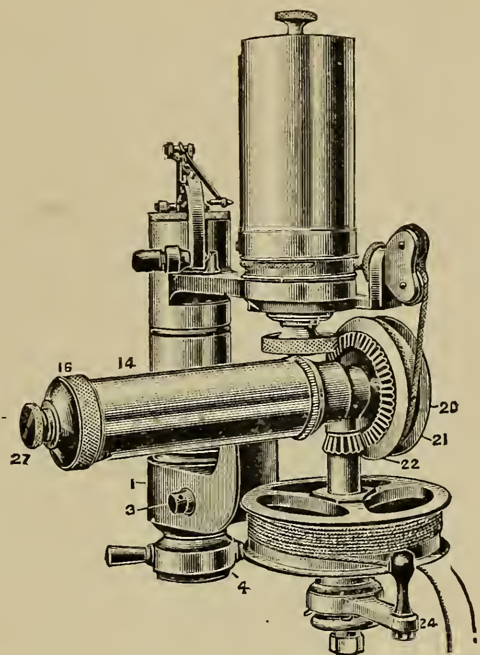


FIG. 39.—Reducing Gear for Crosby Indicator.

88. Reflecting Types of Indicator.—Although the indicator is an important instrument in gas engine work, it does not occupy the position which it holds in steam engine practice where lower speeds and pressures are met with. A much more rapidly moving instrument is needed in internal combustion engines, and such indicators tend to be fragile and are therefore little used except for special investigations. The usual form of such indicator makes use of a beam of light reflected from a vertical mirror which is caused to tilt as the gas pressure rises; at the same time the frame in which the mirror is held is made to move angularly to and fro in time with the motion of the crosshead, thus producing by

the combination of motion the familiar shape of the indicator card. The beam of light, unlike the steel levers of the older form of indicator, has no weight, and therefore no inertia to make it lag behind its true position.

Several models are now in use, but the principle of action is much the same in all. A diagram showing this principle is given in Fig. 40. It consists of a small cylinder containing a piston, just as in the Crosby indicator. The spring, however, is a small straight steel beam, held at the ends, as shown in the diagram (in some instruments the spring is in the form of

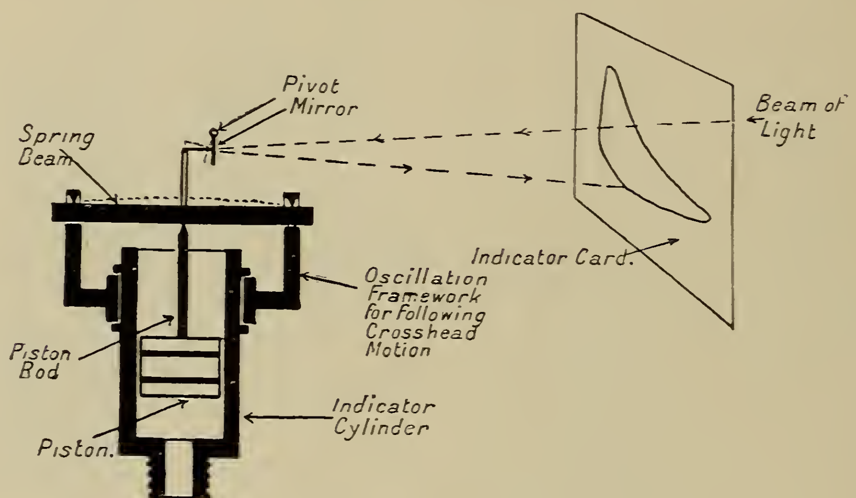


FIG. 40.—Hopkinson Flashlight Piston Indicator.

a diaphragm against which the gas pressure acts without the intervention of a piston). The pressure bends this spring upwards and so tilts the little mirror about a pivot. A beam of light is made to shine on the mirror, and the tilting of the latter deflects the path of the reflected beam through twice the angle through which the mirror is tilted. The reflected beam therefore moves through an angle which is directly proportional to the gas pressure. To give the beam a sideways motion equivalent to the stroke of the engine, the rocking lever is made to reciprocate the bracket carrying the spring beam and the mirror. The mirror therefore gets a partial rotation about a horizontal axis proportional to the pressure and a partial rotation about a vertical axis proportional to

the stroke. This being so, the beam of light on being reflected from the mirror draws a true indicator diagram on the screen, and owing to the speed at which the spot of light moves upon the screen, the diagram will appear as a continuous curve. For permanent record a photographic plate replaces the screen. In this way accurate indicator cards can be obtained even at the highest engine speeds.

Professor Hopkinson claims that with his instrument the indicated horse-power can be measured with an error of less than 1 per cent., whereas in the older forms errors of 5 per cent. or more were common.

89. Mean Effective Pressure.—From an indicator diagram it is easy to find the *mean effective pressure* acting on the piston, i.e., the average pressure in the working stroke less an allowance for the opposing pressures in the idle strokes. There are two methods of doing this :—

(1) A mechanical method, by using the planimeter.
(2) A method of approximate computation, by using what is called the "mid-ordinate rule."

The planimeter is made to follow the pencil line continuously and so automatically to subtract the area due to the lower part of the diagram ; it thus measures directly the area in sq. inches of the closed figure ; if this area be divided by the horizontal length of the diagram in inches, it will give the average breadth of the figure measured vertically, and this, by using the scale of conversion for the indicator spring employed, gives the mean pressure in pounds per sq. inch.*

In the second method a series of equidistant vertical lines (usually eleven in number) are drawn across the diagram as shown in Fig. 41, so that the first and the last just touch the two ends of the diagram. The mid-ordinates bisecting these spaces are then drawn (shown dotted in the figure) ; the lengths included within the diagram of these mid-ordinates are then measured and their sum divided by the number of them.

* In the case of a four-stroke engine it might at first sight seem desirable to divide the average pressure over the *two* outward strokes and not on one of them only, but the accepted convention is that given in the text.

This method gives, as before, the average breadth of the diagram measured parallel to the pressure axis, and so the *mean effective pressure* is obtained.

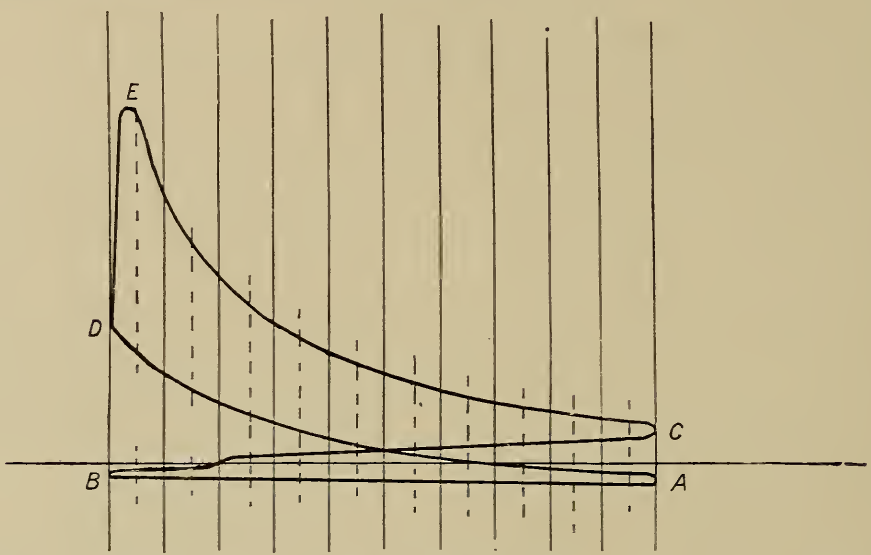


FIG. 41.—Obtaining Area by Mid-ordinate Rule. Area $C\ D\ E$ counts as positive; area $A\ B$ as negative. Mid-ordinate height of negative loop must be subtracted from that of positive loop.

90. Effect of Engine Speed.—The Crosby indicator above described is well suited to gas engines running at normal

FIG. 42.—Indicator Diagram taken from a fast running Engine with a Weak Indicator Spring. When expansion curve is wavy, the I.H.P. cannot safely be determined from the diagram.

speeds, but not to petrol engines, which sometimes run at 2500 r.p.m. or even more. At speeds over 500 the effect

of inertia-lag is felt even in the best of the pencil-indicators. At such speeds the piston has not time to get itself and the rods fastened to it into a position corresponding to that due to a *steady* pressure equal in amount to the momentary gas pressure, and the resulting diagram is therefore incorrect. Moreover, as the engine speed approaches the free-period of the moving parts of the indicator, the latter tends to vibrate sympathetically, and the lines drawn are wavy, as shown in Fig. 42.

91. Estimating indicated from Brake Horse-Power.—Apart from difficulties with indicators at high speeds, it is far from easy to count accurately the number of explosions and to ensure that the "card" shows an average explosion. It is not unusual, therefore; to measure I.H.P. by adding to the B.H.P. figure the B.H.P. necessary to rotate the engine light; this assumes that the mechanical loss is constant at all loads. It is obvious that to assume that the friction at, say, the big and small ends of the connecting rod, or on the piston, will be the same whether there is any thrust in that rod or not, cannot be *strictly* accurate, but Professor Hopkinson has carried out a complete series of tests, using the reflecting type of indicator already mentioned, to elucidate this point. He found that "the difference between indicated horse-power and brake horse-power is rather less than the horse-power at no load under the same conditions of lubrication, mainly because of the difference in the power absorbed in pumping. In the particular engine tested, the error from this cause in obtaining the indicated power would amount to about 5 per cent. The friction is substantially constant from no load to full load, provided that the temperature of the cylinder walls is kept the same, but the influence of temperature is very great." He found the mechanical losses in a 41 H.P. engine to be as follows:—

Suction	3·4	per cent.	of I.H.P.
Piston friction	6·1	,,	,,
Other friction (valve lifting, etc.) .	2·7	,,	,,
Total	12·2	,,	,,

92. Analysis of Motion of Indicator Piston.—It is important to examine mathematically the movement of an indicator when set to observe a very rapidly changing pressure. The piston used in the indicator instrument cannot be absolutely weightless whatever improvement may be made in reducing the weights of the moving levers (either by adopting lighter scantlings or by using a beam of light). Let the pressure acting on the base of the piston of mass M at any time to be p, also let piston area $= a$ and the motion of the piston be S inches for each pound per sq. inch of pressure acting upon it. Then the forces acting on the piston when at a point x above its lowest position are :—

upwards $\qquad p \times a$

downwards $\qquad \frac{x}{\mathrm{S}} \cdot a + \mathrm{M} \cdot \frac{d^2x}{dt^2}$

therefore $\qquad \mathrm{M}\frac{d^2x}{dt^2} + \frac{a}{\mathrm{S}} \cdot x = p.a.$

or

$$\frac{d^2x}{dt^2} + \frac{a}{\mathrm{SM}} \cdot x = \frac{pa}{\mathrm{M}} \qquad . \; . \; . \; (1)$$

Integrate this. The Particular Integral is

$$x = \frac{1}{\mathrm{D}^2 + \frac{a}{\mathrm{SM}}} \cdot \frac{pa}{\mathrm{M}} = \frac{\mathrm{SM}}{a} \cdot \frac{1}{1 + \frac{\mathrm{SM}}{a}\mathrm{D}^2} \cdot \frac{pa}{\mathrm{M}}$$

so that

$$x = \frac{\mathrm{SM}}{a} \times \frac{pa}{\mathrm{M}} = p.\mathrm{S}$$

and the Complementary Function is

$$x = \mathrm{A} \sin\sqrt{\frac{a}{\mathrm{SM}}} \cdot t + \mathrm{B} \cos\sqrt{\frac{a}{\mathrm{SM}}} \cdot t$$

So that the Complete Integral is

$$x = \mathrm{A} \sin\sqrt{\frac{a}{\mathrm{SM}}} \cdot t + \mathrm{B} \cos\sqrt{\frac{a}{\mathrm{SM}}} \cdot t + p.\mathrm{S}$$

Now when $t = o$, $x = o$

therefore $\qquad \mathrm{B} = -p\mathrm{S}$

$$\frac{dx}{dt} = \sqrt{\frac{a}{\mathrm{SM}}} \cdot \mathrm{A} \cos\sqrt{\frac{a}{\mathrm{SM}}} \cdot t - \mathrm{B}\sqrt{\frac{a}{\mathrm{SM}}} \cdot \sin\sqrt{\frac{a}{\mathrm{SM}}} \cdot t$$

and when $t=o$, $\frac{dx}{dt}=o$

therefore $$A=o$$

Substitute these values of A and B and

$$x=p\mathrm{S}\left\{1-\cos\sqrt{\frac{a}{\mathrm{SM}}}\,.\,t\right\}.$$ This means that the piston rises to a height $p\mathrm{S}$ and then oscillates about that position with a frequency equal to $\frac{1}{2\pi}\sqrt{\frac{a}{\mathrm{SM}}}$.

All this assumes, however, that p is a constant or that it increases with such rapidity that it assumes its final value before the indicator piston has had time to move. It would have been more accurate to assume p to rise from zero to its final value in, say, $\frac{1}{n}$th part of a second and to consider what happens during this interval. To do this put $p = a_1.t$ where a_1 has the constant value given by the equation :—final value of pressure$=\frac{a_1}{n}$.

Equation (1) now becomes

$$\frac{d^2x}{dt^2}+\frac{a}{\mathrm{SM}}\cdot x=\frac{a}{\mathrm{M}}\cdot a_1t \quad . \; . \; . \; (2)$$

and the Particular Integral

$$x=\frac{\mathrm{SM}}{a}\left(1+\frac{\mathrm{SM}}{a}\mathrm{D}^2\right)^{-1}\cdot\frac{a_1a}{\mathrm{M}}\cdot t$$

$$=\frac{\mathrm{SM}}{a}\cdot\frac{aa_1}{\mathrm{M}}\cdot t=\mathrm{S}a_1t.$$

Therefore the Complete Integral would be

$$x=\mathrm{A}\sin\sqrt{\frac{a}{\mathrm{SM}}}\,.\,t+\mathrm{B}\cos\sqrt{\frac{a}{\mathrm{SM}}}\,.\,t+\mathrm{S}a_1t.$$

And since $x=o$ when $t=o$

therefore $$\mathrm{B}=o$$

Again $$\frac{dx}{dt}=\sqrt{\frac{a}{\mathrm{SM}}}\,.\,\mathrm{A}\cos\sqrt{\frac{a}{\mathrm{SM}}}\,.\,t+\mathrm{S}a_1$$

but when $t=o:\ \dfrac{dx}{dt}=o$

so that
$$o=\sqrt{\frac{a}{SM}}\,.\,A+Sa_1$$

and
$$A=-Sa_1\sqrt{\frac{SM}{a}}$$

This gives us
$$x=Sa_1t-Sa_1\sqrt{\frac{SM}{a}}\sin\sqrt{\frac{a}{SM}}\,.\,t$$
$$=Sa_1\left\{t-\sqrt{\frac{SM}{a}}\sin\sqrt{\frac{a}{SM}}\,.\,t\right\}$$

Now Sa_1t is height to which piston would rise under the slow static pressure—call it h so that $Sa_1t=h$, and let f be the frequency of the free vibration of the indicator piston. Then
$$f=\frac{1}{2\pi}\sqrt{\frac{a}{SM}}\ \text{or}\ 2\pi f=\sqrt{\frac{a}{SM}}$$

so that
$$x=h-\frac{Sa_1}{2\pi f}\sin\sqrt{\frac{a}{SM}}\,.\,t$$
$$=h-\frac{h}{2\pi ft}\sin 2\pi ft$$

or
$$x=h\left\{1-\frac{1}{2\pi ft}\sin 2\pi ft\right\}\quad .\ .\ .\ (3)$$

This means a fractional lag of $\dfrac{1}{2\pi ft}$ as a maximum, but for any particular case it can be calculated thus. We may put f as 300, which about represents the use of an instrument of the Hopkinson type.

Then
$$x=h\left\{1-\frac{1}{1890t}\sin 1890t\right\}$$

It will be useful to compute a few values for this for cases in which the value of t is much shorter than the periodic time of the instrument. When this is so sin $1890t$ can be written with sufficient accuracy as

$$(1890t)-\frac{(1890t)^3}{6}$$

or
$$x=h\left\{1-1+\frac{(1890t)^2}{6}\right\}$$
$$=h\,.\,\frac{(1890t)^2}{6}=600{,}000ht^2.$$

The relation between x and t in these early stages is therefore parabolic. The time t starts, so to speak, first, but x soon increases and gradually catches up.

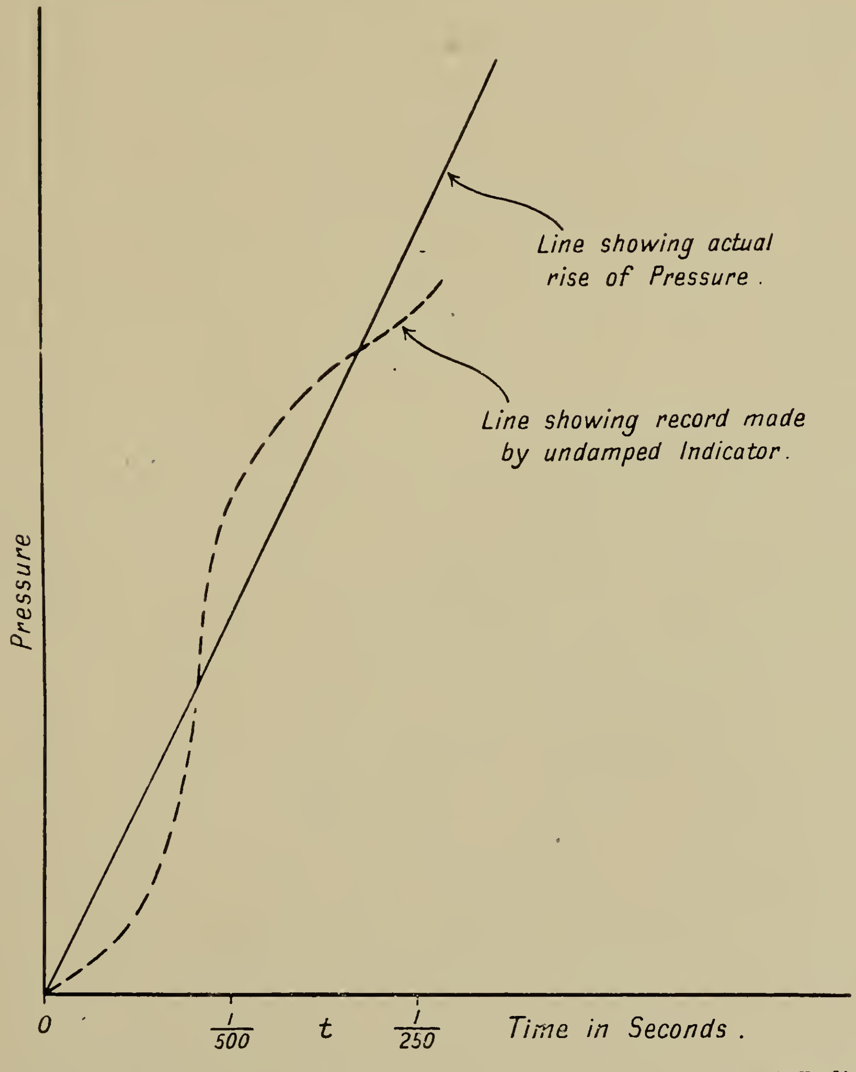


FIG. 43.—Diagram illustrating the way in which an undamped Indicator would follow a rapid explosion. Period of Indicator, $\frac{1}{300}$ sec.

Thus for $t=\frac{1}{100000}$ sec.

$$\frac{x}{h}=\frac{600.000}{(100.000)^2}=0\cdot00006$$

For $t=\frac{1}{10000}$ sec.

$$\frac{x}{h}=0\cdot006$$

for $t=\frac{1}{1000}$ sec.

$$\frac{x}{h}=0\cdot6$$

but for this value of t our approximation no longer holds. For $t=\frac{1}{1000}$ the calculation should proceed thus

$$\frac{x}{h}=\left(1-\frac{1000}{1890}\sin 1\cdot890\right)$$

$$=1-\frac{1}{1\cdot89}\sin 108^{\circ}$$

$$=1-\frac{1}{1\cdot89}\times0\cdot95=1-0\cdot50$$

$$\therefore\frac{x}{h}=0\cdot50$$

showing that the instrument is beginning to pick up. Evidently therefore it will not do to use an instrument for recording an explosion occurring in $\frac{1}{1000}$ sec., unless its own frequency exceeds 300.

The following table shows a series of values, and in Fig. 43 they are shown plotted.

t secs.	$1890t$	$\sin 1890t$	$\left(\frac{1}{1890t}\sin 1890t\right)$	$\frac{x}{h}$
$\frac{1}{100000}$	—	—	—	0·00006
$\frac{1}{10000}$	—	—	—	0·006
$\frac{1}{2000}$	—	—	—	0·15
$\frac{1}{1000}$	1·89	0·95	0·50	0·50
$\frac{1}{500}$	3·78	−0·60	−0·16	1·16
$\frac{1}{400}$	4·72	−1·00	−0·21	1·21
$\frac{1}{250}$	7·56	0·96	0·13	0·87
$\frac{1}{100}$	18·9	0	0	1·00

Whenever t is a multiple of $\frac{1}{600}$ the value of $\frac{x}{h}$ will be 1·00. The above curve does not of course take account of the frictional forces which prevent the indicator piston continuing to vibrate indefinitely. Students are recommended to work the problem out, introducing into equation (2) a term representing the frictional force. The result will be to multiply the oscillatory term by a factor of the type ϵ^{-qt} which, when the student has plotted the resulting curves, will show that the straight line is soon followed once the curve comes up to and crosses it. From the curve in Fig. 43 it is clear that for recording an explosion occurring in $\frac{1}{10000}$ sec. this indicator with its $\frac{1}{300}$ period would be inadequate. The piston would scarcely have moved. For an explosion occupying ten times as long, i.e. $\frac{1}{1000}$ sec., the indicator would still be lagging a long way behind. For a $\frac{1}{300}$ sec. explosion the actual maximum pressure would be very fairly represented, but not the shape of the explosion wave. In fact for useful readings the instrument should not be used for any sharper explosion than $\frac{1}{200}$ sec. For an ordinary gas engine explosion occurring in $\frac{1}{100}$ sec. the instrument would be quite satisfactory.

93. Engine Tests.—These are the tests applied when the engine has been constructed and built. They consist in the actual running of the engine as nearly as possible under working conditions. The longest run is at full load, and it is customary afterwards to run for a time at half load and at no load. Sometimes it is specified that runs should be made at three-quarter load and one-quarter load, and occasionally overload tests are made. It is almost impossible for one observer to do all that is necessary in such a test. The workshop custom is to measure —

(1) The brake-horse-power.
(2) The amount of fuel used per hour.
(3) The quantity of cooling water used per hour.
(4) The rise in temperature of cooling water between inlet and outlet.
(5) The revolutions per minute.

Sometimes the indicated horse-power is also measured, but

with high-speed engines this is not usual. Two observers can carry out the above tests. When tests are made not as a matter of workshop routine but as a matter of special research in workshop or college laboratory a very large number of additional tests are carried out and many observers are needed.

Indicated horse-power is deduced from the indicator diagram, obtained in the manner already described, and the record of the number of explosions per minute. The work done in ft.-lb. in a working stroke is

Mean effective pressure in lb. per sq. inch $\times$ area of piston in sq. inches $\times$ length of stroke in feet.

This, multiplied by the number of working strokes per minute and divided by 33,000, gives the I.H.P. This is, of course, the same as the formula $\frac{\text{P.L.A.N.}}{33,000}$ given in books on the steam engine, but it must be observed that N must be taken as the number of working strokes per minute, not the number of revolutions per minute, i.e. in a four-cycle engine N cannot be more than half the number of r.p.m. and may be much less if the governor should be cutting out explosions.

Brake horse-power is the power exerted at the crank-shaft and it is measured by applying a frictional load to the flywheel, as shown in Fig. 44. A test of this kind is often called a "bench test." A number of wooden blocks are fitted loosely to the rims of the flywheel and connected together by one or more ropes or by a canvas belt. A number of heavy weights are hung as at P_1 and a spring balance is placed at P_2. The distance D is measured in feet. As the force due to the weights P_1 is greater than the force at P_2 there will be a force acting against the direction of the arrow of $P_1 - P_2$. The work done therefore by the flywheel in one revolution in the direction of the arrow = force $\times$ distance moved

$$=(P_1-P_2)\times \pi D,$$

and if there be N revolutions per minute the

$$\text{B.H.P.}=\frac{(P_1-P_2)\times \pi D\times N}{33,000}.$$

As this power is all being spent in friction it produces heat,

and in the larger engines the flywheel has to be cooled by a water spray. In the very biggest engines even this is not enough and a special water churning brake is used, or else the load is "taken up" electrically. In the latter case, the engine drives a dynamo and the electrical output of the dynamo is measured in kilowatts, when if the efficiency of the dynamo be known the B.H.P. can be deduced by turning the K.W.

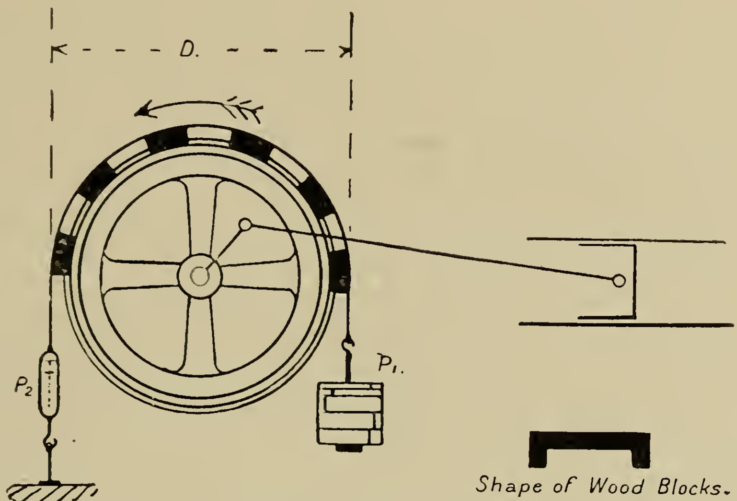


FIG. 44.—Arrangement for Brake Horse-Power Test.

into H.P. and dividing the result by the dynamo efficiency.

94. Mechanical Efficiency.—The ratio of $\frac{\text{B.H.P.}}{\text{I.H.P.}}$ is the mechanical efficiency of the engine. It is usually from 75 per cent. to 85 per cent. at full load. The amount by which the I.H.P. exceeds the B.H.P. measures the power lost in engine friction. It is nearly independent of the load.* Thus in a special test of an engine of 76 I.H.P. the engine friction was found to be

10·8 H.P. at no load
11·0 H.P. at $\frac{1}{2}$ load
11·7 H.P. at full load

in each case the r.p.m. being nearly 200.

* *See* p. 131.

These figures show that engine friction is almost independent of the load. It rises however very rapidly with rise in speed.

95. Measurement of Cooling Water.—From time to time during the test a measuring vessel can be placed under the outlet, or in some other convenient way a measurement be made of the number of gallons of water passing through the water jacket per hour. A thermometer is placed in the inlet and another in the outlet. Then the rise in temperature of the water multiplied by the number of *pounds* of water flowing per hour gives the number of heat units carried away by the cooling water per hour.

96. Heat in Exhaust Gases.—The record of the test will show the amount of fuel used per hour. Its calorific value is either known in advance or is measured at the time. This gives the number of heat units supplied per hour. Some of these are turned into work (B.H.P.); others are lost in engine friction (I.H.P. minus B.H.P.); others are carried away in the cooling water; and the residue are carried away in the hot exhaust gases. As it is not very easy to measure the quantity and temperature of the exhaust gases, it is customary to ascertain the amount of this loss by a subtraction sum. The B.H.P. expressed in heat units per minute is added to the heat units lost per minute in engine friction and to the cooling water loss per minute. Then this total is subtracted from the heat units supplied to the engine per minute, and the result is called the exhaust loss.

The engine friction itself produces heat, some of which may find its way into the cooling water. Again, the exhaust gases in passing out of the cylinder often come into contact with a portion of the water jacket and give heat to the cooling water. Both tend to exaggerate the true cooling water loss, but their effect is not considerable.

A suitable test report form for entering up these measurements is reproduced opposite.

97. Heat Balance-sheets.—A heat balance-sheet as applied to a gas engine is a statement of the way in which the total amount of heat passed into the engine is employed. In the

No. of engine ....
Dia. of cylr.........................
Stroke ....
Revs. per min. ....
Cyclic irregularity (when known) .....
Rated B.H.P. ....
Area of producer grate ....
Nature of vaporiser ....
,, governor ....

	Quantity of water used in			Remarks. (State nature of load applied and of ignition arrangements.)
Gallons per min.	Producer.	Scrubber.	Max. pressure from indicator card.	
(16)	(17)	(18)	(19)	(20)

%
100

Compression ratio ....

"Air standard" of efficiency ....

Ratio $\frac{\text{indicated thermal efficiency}}{\text{air standard efficiency}}$ ....

................Observer.

Date...............................

No. of engine ......
Dia. of cylr. ......
Stroke ......
Revs. per min. ......
Cyclic irregularity (when known) ......
Rated B.H.P. ......
Area of producer grate ......
Nature of vaporiser ......
,, governor ......

INTERNAL COMBUSTION ENGINE TEST REPORT.

Report No. .... *Maker of Engine* .... *Maker of Producer* ....

FUEL CONSUMPTION TEST. *Nature of Fuel used during test* .... *Cal. Value* .... *C.H.U. per*

Duration of test.	Revs. per min.	Explosions per min.	Air. Temp.	Air. Pressure ins. of mercury.	Gas at Inlet. Temp.	Gas at Inlet. Pressure above or below atmosphere ins. of water.	Gas at Inlet. Calorfic value C.H.U. per standard cu. ft.	Quantity of fuel used. Cu. ft. of gas suppleid to engine (meter reading).	Quantity of fuel used. Pounds of fuel used in producer.	B.H.P.	I.H.P.	Mechl. effy.	Jacket water. Temp. at inlet.	Jacket water. Temp. at outlet.	Jacket water. Gallons per min.	Quantity of water used in Producer.	Quantity of water used in Scrubber.	Max. pressure from indicator card.	REMARKS. (State nature of load applied and of ignition arrangements.)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)

GOVERNOR TEST.

	Revs. per min.	% variation from full load steady speed.
Load on, steady speed		
Load off, momentary maxm. speed. . .		
,, ,, steady speed		
Load on, momentary min. speed . . .		

HEAT ACCOUNT.

(Full Load.)

	C.H.U.	%
Thermal equiv. of work done per min. . . .		
Loss due to engine friction ,, ,, ...		
Heat loss in jacket ,, ,, ...		
,, ,, ,, exhaust ,, ,, ...		
Heat received ,, ,, ... (fuel per min. × Ca . Va .)		100

Compression ratio ....

"Air standard" of efficiency ....

Ratio $\frac{\text{Indicated thermal efficiency}}{\text{air standard efficiency}}$ ....

................Observer.

Date..............................

early days of gas engine work it was easy to remember that roughly—

Heat passed to water jacket . .	40	per cent.
Heat left in exhaust gases . .	40	,,
Heat converted into work . .	20	,,
	100	,,

In the experiments made by the Institution of Civil Engineers' Committee the full-load heat balance-sheet was given as :

Designation of Engine.	L.	R.	X.
Exhaust waste	35·3	40·0	39·5
Jacket waste	23·5	29·3	25·0
Radiation	7·6	10·0	7·3
B.H.P.	26·7	28·3	29·8
Total	93·1	107·6	101·6

In these experiments the exhaust waste was measured by passing the exhaust gases into a water-jet calorimeter. Jacket waste was measured as the product of quantity of cooling water passed and rise of temperature. Radiation includes engine friction as well as radiation proper. B.H.P. was measured by a rope brake.

Engine L shows a deficit in the total, so that there must have been some error in the experiments. Dugald Clerk in his paper * before the Institution of Civil Engineers, "On the Limits of Thermal Efficiency in Internal Combustion Motors," endeavoured to correct this measurement from several different possible points of view. He also extended the same treatment to tests R and X in order to get the true balance-sheet, and putting in I.H.P. instead of B.H.P. (the Committee's records were complete enough to permit of this), he found :—

* February 26, 1907.

Designation of Engine.	L.	R.	X.
Exhaust waste	41·0	37·1	39·9
Jacket waste and radiation . .	27·2	29·6	25·4
I.H.P.	31·8	33·3	34·7
Total	100·0	100·0	100·0

Clerk then points out that the 27·2 per cent. of jacket waste and radiation for test L is obviously too low, and that heat appears to have been lost in some way. He therefore took the total of the exhaust waste and jacket waste and radiation items, i.e. 68·2 per cent. and attributed 34·1 per cent. to each, so making the balance sheet into :—

Designation of Engine.	L.	R.	X.
Exhaust waste	34·1	37·1	39·9
Jacket waste and radiation . .	34·1	29·6	25·4
I.H.P.	31·8	33·3	34·7
Total	100·0	100·0	100·0

Clerk considered this balance-sheet probably represented the distribution of heat in the engines more accurately than either of the others.

These various attempts at a heat balance-sheet have been given in order to show how very difficult it is to obtain a really accurate statement. The exhaust wastes originally given for L, R and X were 35·3 per cent., 40·0 per cent., and 39·5 per cent., and have now become 34·1 per cent., 37·1 per cent. and 39·9 per cent.

But the matter does not end even here, as Clerk brought into use the values found by him for the specific heat—values which showed a marked increase with rise of specific heat—and used them in some separate experiments of his own with

the engine X used by the Committee. He then found that the balance-sheet became :—

Heat-flow during explosion and expansion	16·1 per cent.
Heat contained in gases at end of expansion	49·3 ,,
I.H.P.	34·6 ,,
	100·0

Compare this with the balance-sheet given on p. 142 based on the Committee's experiments :—

	Committee's Trials.	Mr. D. C's. Trials.
Heat-flow during explosion and expansion .	25·4	16·1
Heat contained in gases at end of expansion	39·9	49·3
I.H.P.	34·7	34·6
Total	100·0	100·0

The discrepancies shown here are indeed serious. Clerk's comment on them is as follows: "The indicated work is practically the same in both trials and the sum of the other two items is the same also, but the distribution is different. Less heat flows through the cylinder-walls as determined by the author's (Mr. Clerk's) new method, and the exhaust gases contain more heat than the Committee's calorimeter trials show. The ordinary trials show 9·3 per cent. too much heat as passing through the cylinder-walls, and practically the same amount too little appears in the exhaust calorimeter. That is, 18·8 per cent. of the total heat remaining in the hot gases at the end of the expansion passes into the cylinder water-jacket during the flow through the exhaust valve upon the first opening and while the piston is making its exhaust stroke. This seems to be a quite reasonable portion of the total heat, such a portion as experience would lead one to expect. These new diagram trials afford, in the

author's (Mr. Clerk's) view, a more accurate heat-distribution balance-sheet than has yet been obtained in any engine, from which can be deduced the ideal efficiency of the working fluid. Adding together

Heat contained in gases at end of expansion .	49·3
I.H.P.	34·6
	83·9

Then $\frac{34 \cdot 6}{83 \cdot 9} = 0 \cdot 41$. That is, if this balance-sheet be correct and the heat loss be assumed as entirely incurred at the beginning of the stroke, then the maximum efficiency of the actual working fluid for the compression and expansion is 41 per cent.* of the total heat supplied."

Even with such very considerable discrepancies in the heat balance-sheets as those discussed above, the student will none the less remark that the heat utilized has now grown from about 20 per cent. to well over 30 per cent. This all-important improvement has occurred therefore in spite of the many uncertainties as to how the lost heat divided itself up. It is indeed one of the fortunate features of gas engine manufacture that improvements do not have to attend the settlement of the many intricate problems with which gas engine operation is bound up, but proceed by the trial and error of experiment with such guidance as theoretical considerations have been able to afford. The great want which in the past caused so much theoretical difficulty was accurate knowledge of the values of the specific heats of the working fluids.

98. Engine Tests.—(*a*) The following figures are taken from a test on a 200 H.P. engine and suction plant by Mathot.† The engine was of the four-cycle double-acting type and was tested at the works of the well-known firm of Gasmotoren Fabrik, Deutz-Cologne.

Piston diam.	$21\frac{1}{4}$ in.
,, stroke	$27\frac{9}{16}$ in.

* The "Air standard" efficiency for this engine = 0·49; the "Gas standard" of efficiency would (see p. 85) be 81 per cent. of this or 0·40, which is very near the figure above given.

† I.M.E., 1905.

FULL LOAD TESTS.

	1904.	
	March 14.	March 15.
Average r.p.m.	151·29	150·20
B.H.P.	214·22	222·83
Duration of test, hours	3	10
Average temperature of water after cooling piston	117·5° F.	—
Average temperature of water after cooling cylinder and valve seats . . .	135° F.	—
Water consumption for cooling piston, gallons /hour	39	—
Water consumption per hour in vaporizer (anthracite fuel), galls. /hour . . .	—	14·2
Water consumption per hour in scrubbers, galls. /hour	—	318
Average temperature of gas at outlet of generator	—	558° F.
Average temperature of gas at outlet of scrubbers	—	62·5° F.
Gross fuel consumption per B.H.P. hour	0·727 lb.*	0·720 lb.
Corresponding Thermal efficiency	19 per cent.	24·4 per cent.

Other interesting figures are—

Water consumption in galls. per B.H.P.-hour—

1. For cooling cylinder, stuffing boxes, valve seats and jackets . . . . . . 4·65
2. For cooling piston and piston rods . . 1·75
3. For vaporizer . . . . 0·0655
4. For washing the gas in the scrubbers. . 1·42

Also :—

Water converted into steam per lb. of fuel consumed in generator . . . . 0·193 galls. or 1·93 lb.

(*b*) In a careful test carried out by J. T. Nicolson on a Crossley gas engine and suction producer plant, the calorific value of the gas was 156·5 B.Th.U. as determined by analysis, and 149 B.Th.U. per cubic foot by Junker's calorimeter at

* Includes fourteen hours of fires banked up.

the temperature and pressure of the calorimeter. The following measurements were made :—

B.H.P. = 559.

Gas per hour = 29,037 cu. ft. corrected to 0° C. and 760 mm.

Gas per B.H.P. = 51·94 cu. ft.

Heat supplied = 51·94 × 156·5 = 8,128 B.Th.U. per B.H.P.-hour Brake thermal efficiency $= \frac{1{,}980{,}000}{778} \times \frac{1}{8{,}128} = \frac{2{,}546}{8{,}128} =$ 31·3 per cent.

Variation in engine speed when horse-power was instantaneously dropped from 600 to 50 was from 119·4 to 121·4 r. p.m., corresponding to a total variation of $1\frac{2}{3}$ per cent. of mean speed. No back-firing was observed to take place when this was done. These tests show remarkably good thermal efficiency and satisfactory closeness of governing.

(*c*) A third trial is that of a **150 B.H.P. six cylinder vertical gas engine** which was run for six hours on full load. The gas was taken from a pressure producer and had its calorific value measured every hour by a Simmance-Abady Calorimeter. Readings were taken every half-hour of the B.H.P.

Average air temperature . .	72·2° F.
Average air pressure . .	29·56″ Hg.
Cu. ft. gas used per hour .	13,000.
Average Calorific value (lower).	128·1 B.Th.U. per cu. ft.
Engine speed	325 r.p.m.
B.H.P.	151·3.

B.Th.U. consumed by engine per B.H.P.-hour = 10,590, showing a brake thermal efficiency of $\frac{1{,}980{,}000}{10{,}590 \times 778} = 24{\cdot}1$ per cent.

99. Governors.—The most usual type of governor has two balls fastened by arms to the shaft and rotating with it. Such governors are shown in Figs. 45 and 46. In the former figure it is arranged on a vertical shaft, and in the second on a horizontal one. In Fig. 45 A is the shaft; it carries on it the bracket C_1C_2, and at C_1 and C_2 are hinged the arms D_1 and D_2 carrying the balls B_1 and B_2. As the shaft rotates the balls tend to fly outwards by centrifugal force, and in doing so lift the

sliding collar F. To this collar can be attached a lever or a system of levers to act on the engine. The way in which they act will be discussed later. In Fig. 46 the same sort of action occurs. As the balls B_1 and B_2 fly out they carry with them the links H_1 and H_2 which are hinged at C_1 and C_2 to the arms of a bracket G fixed on the shaft. This outward movement causes the sliding collar F to slide towards the fixed collar D, and so to compress the spring E. The faster the shaft A rotates the more the spring is compressed.

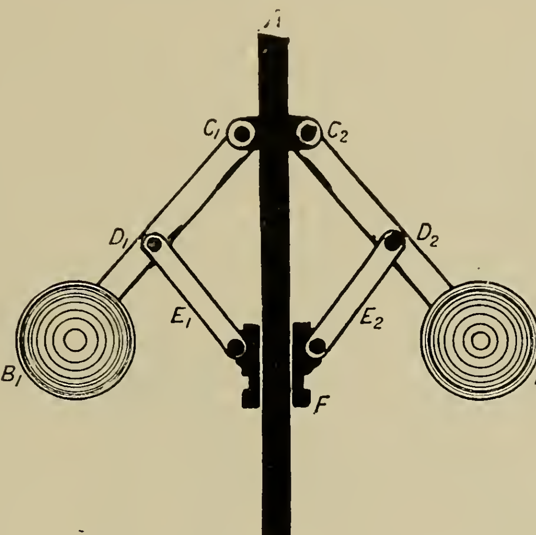


FIG. 45.—Vertical Governor.

In the vertical governor the effort of the balls to fly out is balanced against gravity. In the horizontal form it is balanced against the force exerted by the spring. The horizontal governor is most used with internal combustion engines.

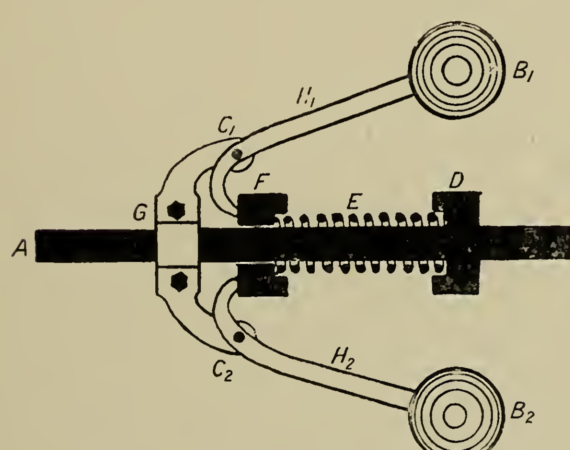


FIG. 46.—Horizontal Governor.

100. Method of Controlling Engine.—The motion of the sliding sleeve of either the horizontal or vertical governor causes a lever to be moved which controls the engine and brings it back to its correct speed. This lever can be used to—

(1) Cut off the whole of the fuel supply for one or more strokes ; or

(2) Reduce the amount of fuel used per cycle, leaving the air supply untouched; or

(3) Reduce the amount of both fuel and air keeping the proportion of fuel to air the same; or

(4) Cause the ignition to come later, or be cut off altogether.

101. Hit and Miss Governing.—The first of these alternatives is known as "hit-and-miss" governing, because when the speed gets too high the governor lever is made to lift up a small piece of metal which lies between the gas valve tappet and the valve stem. This is shown in Fig. 47. B is the piece of metal in question. When the cam E pushes the roller G it makes the tappet rod GDC, which is pivoted at D, push the valve A open. (The valve A is the gas admission valve; after the gas passes this valve it joins the air supply, and both pass through a larger valve, which opens a little earlier, into the cylinder.) This could not be done if B were not in line between C and A as shown. The governor lever lifts B out of this line when the speed is too high, and the valve A is consequently not opened and no gas reaches the cylinder. This reduces the engine speed until the governor again inserts the piece B. The engine speed will therefore be kept steady at all loads up to the maximum load the engine can take.

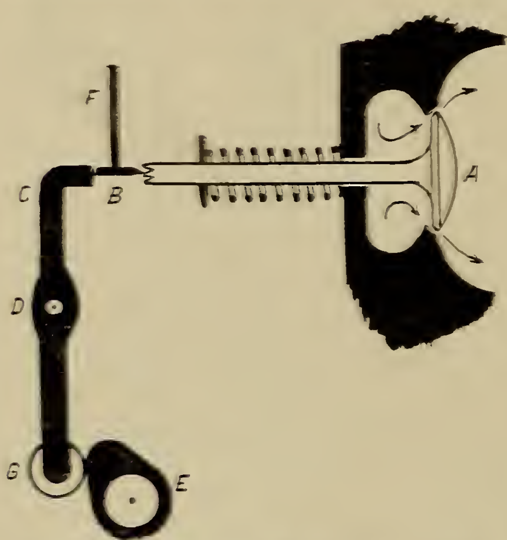


FIG. 47.—Diagram illustrating "Hit-and-Miss" Mechanism.

The effect of hit and miss governing is clearly shown on the indicator diagram. In Fig. 48 is shown such a diagram taken during two successive strokes of an engine. The "hit" or working stroke is shown at ACD, and the "miss" or idle stroke is shown at AB. The compression line of "miss" stroke lies below that of the "hit." This is because the gas valve is closed on the suction stroke and the air valve acting

alone is not large enough to fill the cylinder without throttling the entering air, with the consequence that the " miss " suction pressure line lies below that of the " hit " suction. It will be noticed moreover that the compression line AB almost coincides with the expansion line BA. In order to exhibit better what was going on during the two suction strokes a " suction " diagram was taken. It is shown in Fig. 49. A suction diagram is one taken with a very weak spring and with a stop fixed above the indicator piston to limit its rising above a point corresponding to about 3 lb. per sq. inch. The " hit " and the " miss " suction strokes both start at A, but the latter lies at AB much below the former at AD. The beginning of the compression strokes are respectively BC and DE. FA is the exhaust line after explosion, and it is seen to fall below the

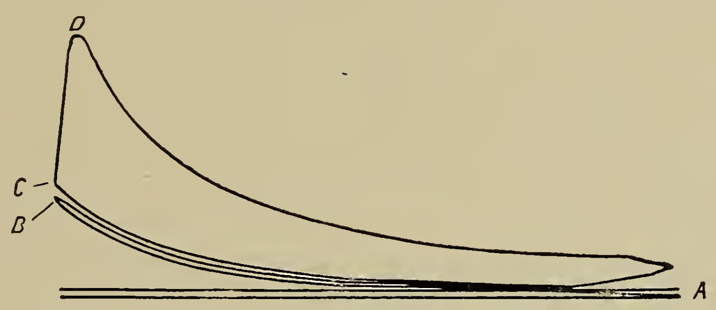


FIG. 48.—Indicator Diagram, from a 25 h.p. gas engine, showing effect of " Hit and-Miss " Governing.

atmospheric line ; this shows the very useful " scavenging," or clearing out, effect of the rush of exhaust gases along the exhaust pipe.

102. Exhaust Governing.—The second of the alternatives mentioned in paragraph 100 is called quality governing, as the governor is here used to reduce the working charge in richness or quality. It does this by *partly* cutting off the fuel supply. This reduces the mean thermal value of the charge admitted, with the result that the pressure produced is less and the engine speed falls to its normal value. The disadvantage of this method of governing is that the composition of the mixture is continually varying, and it is not possible to find any fixed ignition point to suit all mixtures. Moreover if the mixture be made too weak it will not fire at all.

103. Quantity Governing.—This is the method of the third of the alternatives given in paragraph 100. It consists in throttling the working charge, of air and gas, as it is about to enter the cylinder. The consequence is that at the end of the suction stroke only a part of the usual charge has got into the cylinder and the pressure is therefore less than atmospheric and the compression pressure is much reduced. The low compression pressure means a low explosion pressure and the engine speed therefore falls back to its normal amount. In this method of governing, as in those previously mentioned,

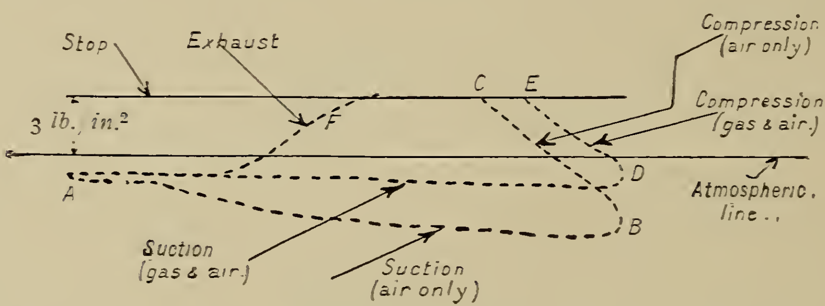


FIG. 49.—" Suction " Diagram from same engine as Fig. 48.

the compression ratio is unaffected, and the "air standard" of efficiency is also unaltered. Quantity governing is coming more into force. At one time it was only used on Continental engines, but now is very general in large engines made in England and America.

104. Retarding Ignition.—This means of altering engine speed, though theoretically feasible, is not practical. To make the spark come later in the expansion stroke, instead of at the very beginning of it, is, of course, to diminish the average working pressure and therefore the speed of the engine. But as it leads to the use of just as much fuel whether the engine is giving a full power stroke or not, it is uneconomical, and as moreover there is the risk of some of the charge getting into the exhaust unburnt it may lead to unpleasant explosions in the exhaust pipe. Cutting off the ignition altogether introduces the same disadvantages in an aggravated form.

105. Present Practice.—Generally speaking large gas engines usually have *quantity governing*, whilst small ones very com-

monly have "*hit-and-miss*." Motor-car engines are usually governed by hand on the throttle, so producing an irregular sort of quantity governing. An objection to the "hit-and-miss" system is that in order to produce a reasonable measure of uniformity of angular velocity in the crank shaft a very heavy flywheel becomes necessary. This adds to the cost of the engine and diminishes its mechanical efficiency. It may in fact be said that the two merits which have enabled the "hit-and-miss" gear to be used as much as it is, are its great mechanical simplicity and its ability to keep constant the proportions of gas and air in the incoming charge, so enabling the engine always to be run on its most economical mixture. Continental makers were the first to break away from this system of governing, by arranging that the governor should produce a variable lift of the gas valve by means of a conical cam. As the air supply was not interfered with this meant a continually changing richness of charge and hence a corresponding change in thermal efficiency; this is well shown in Fig. 26 on p. 87. The tendency now is towards a regulation of both the gas and the air supplies by throttling them after mixture, with the advantage that the mixture being of constant composition the rate of ignition does not vary with the varying position of the governor. Present practice has not settled down to any definitely accepted standard, and there are few matters relating to the gas engine which are the subject of more patents.

106. Turning Moment at Crank-Shaft.—The pressure exerted on the piston during the explosion stroke falls rapidly as the gases expand. Thus the total force exerted on the piston is far greater at the early part of the stroke than it is at mid-stroke and still greater than it is near the end of the stroke. Also the angle between the engine centre line and the connecting rod is continually changing; as is the perpendicular distance from the crank-shaft centre to the line of the connecting rod. This leads to a very irregular turning effort at the crank-shaft.

Thus in Fig. 50, if F be the force on the piston, P the force actingalong the rod, and R the reaction from the cylinder wall, R and P can be calculated when F is known by applying the

triangle of forces as shown; thus P = F cosec BCA. The turning moment about the crank-shaft centre A

$$= P.\ AB.\ \sin ABC = F.AC.$$

This is the torque that causes the engine to do work.

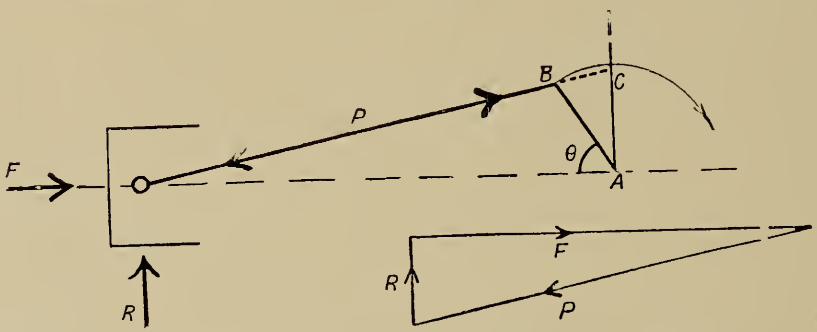


Fig. 50.—Forces acting at Gudgeon Pin and at Crank Pin.

In order to study it more fully it is necessary to consider the variation of F during the cycle. Now F is proportional to the height of the indicator diagram, and if the four successive strokes of a four-stroke engine are set out in one straight

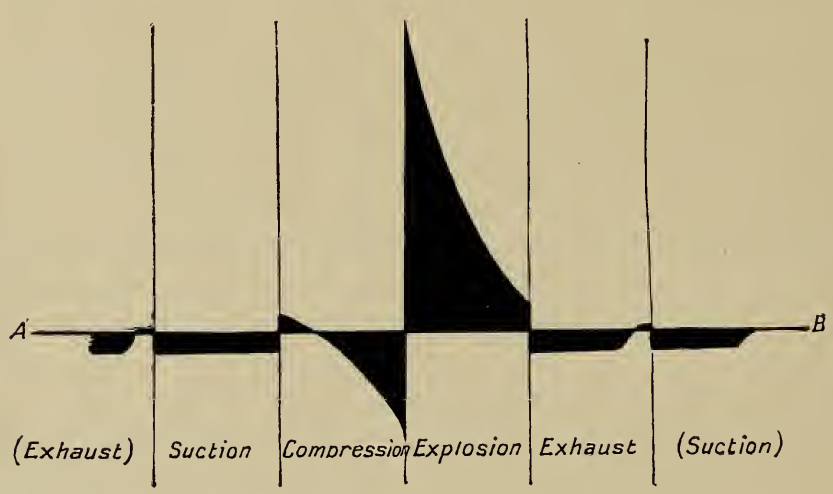


Fig. 51.—Effort tending to Rotate Crank in each of the Four Strokes. (Pressures opposing motion of crank-pin are shown as negative.) For indicated diagram represented, see Fig. 41.

line and the indicator diagram lines are carefully reproduced, a diagram of the form shown in Fig. 51 is obtained. Care must be taken to allow for the *sign* of the pressure, e.g., in the

portion of the curve corresponding to compression, the piston is retarding the motion of the crank-pin and F is negative; that portion of the curve is therefore placed below the line AB in Fig. 51. The diagram repeats itself after every four strokes. It is obvious from Fig. 51 that the force F acting on the piston varies very greatly during the complete cycle. To remedy this it is possible to have two cylinders acting on the same crank-shaft and to put the cranks at an angle of 180° to one another. This means that while the first engine is on its explosion stroke the second will be on either its exhaust or compression strokes, so that the crank-shaft will get two impulses every four strokes instead of only one impulse. This may be further improved by having four

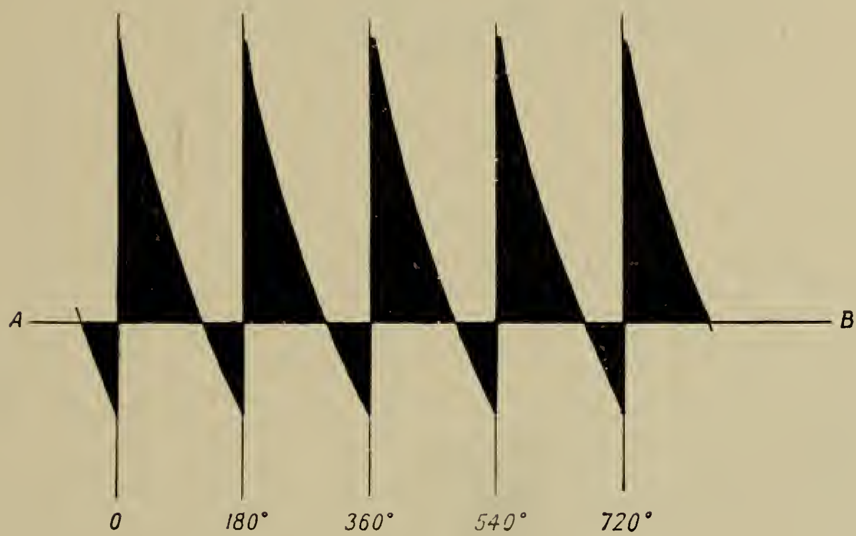


FIG. 52.—Same as Fig. 51, but with Four Cylinders. (Cranks at 180°.)

cylinders so that there is an explosion in every half revolution and the net value of F is as shown in Fig. 52. Even this curve is very saw-toothed, but it gives a steadier forward effort on the crank-pin than does Fig. 51.

107. Inertia Forces.—The curve of Fig. 52 is also subject to a further influence due to the inertia of the moving parts. If the engine rotated very slowly this correction would be of little importance, but at ordinary engine speeds it has considerable influence. At the beginning of the explosion stroke the piston is at rest, and by mid-stroke it may be moving at 2000

ft. per min. This means a considerable acceleration and therefore calls for an equally considerable *force* to bring it about. Some of F, therefore, is required in overcoming the inertia of the piston and small end of the connecting rod before any force can be transmitted to the crank-pin. This force is not lost, but reappears as the piston slows down in the second half of the stroke. The effect of inertia is therefore to lower the peaks of Fig. 52 to an amount depending on the engine speed and to raise the valleys by a similar amount. This is a useful effect, as it tends to minimise the saw-toothedness and

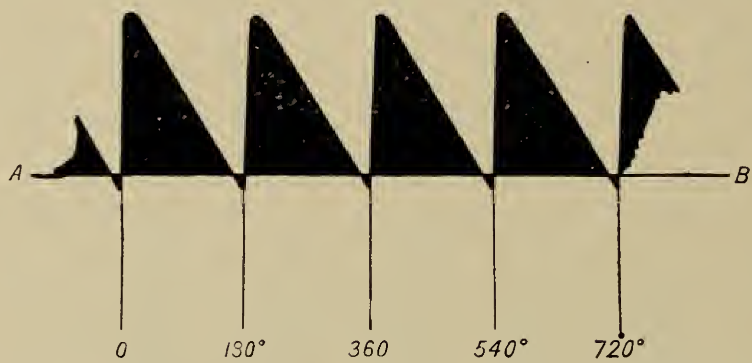


FIG. 53.—Same as Fig. 52, but with allowance for Inertia of Reciprocating Parts.

change it into a curve similar to that in Fig. 53. The amount of this change depends on the speed of rotation, and varies in proportion to its *square*.

108. Flywheel Effect.—The kinetic energy stored up in a flywheel is calculated from the following formula or one derived from it.

$$\text{KE.} = \tfrac{1}{2}\text{I}\omega^2 \text{ ft.-lb.}$$

where I = moment of inertia about the axis of revolution and ω = angular velocity in radians per second.

From this it follows that

$$\frac{d}{d\omega}(\text{KE}) = \text{I}\omega.$$

For a small variation in ω compared with KE it is therefore necessary that either I or ω should be large. For the ordinary purposes of industry it is sufficient to ensure that the range

of angular velocity never exceeds by more than $\frac{1}{25}$th to $\frac{1}{30}$th part the mean speed. For the driving of continuous current generators only half the above variations are permissible, whilst for the driving of alternators in parallel the requirements are far more stringent, involving a permissible speed variation of but $\frac{1}{150}$th or even in some cases $\frac{1}{200}$th part of the mean. Mathot * has suggested the following formula for use in calculating the dimensions which should be given to flywheels of different types of Otto cycle engines :—

$$P = k\frac{N}{D^2.a.n^2}$$

where

P = weight of rim (without arms or boss), in tons.
D = diameter to centre of gravity of rim, in feet.
a = degree of cyclic irregularity permissible.
n = revolutions per minute.
N = B.H.P.
k = coefficient determined as below.

For single cylinder, single acting . . . k = 475,000
For two opposed cylinders, single acting, or one cylinder double acting . . . . k = 300,000
For two cylinders, single acting, tandem or twin . . . . k = 225,000
For two cylinders, double acting, tandem or twin k = 75,000

Note.—Total weight of flywheel may be put as 1·4 P.

109. The usual way to speak of **cyclic irregularity** is that above described.† It amounts to defining cyclic irregularity as the fraction which the range of instantaneous angular velocity bears to its mean value in any one complete cycle. There is, however, another way of considering the matter. Thus Mr. L. Schüler in a paper dealing with the driving by gas engines of alternators operated in parallel remarks : " The speed of the machines should be as uniform

* *Internal Combustion Engines* (1910).

† This is a more accepted way than that described in the first edition of this book.

as possible and should in any case be such that the amplitude of the angular oscillation does not exceed two electrical degrees." An electric degree means of course $\frac{1}{360}$th part of the angular distance, the passage of which by the armature corresponds to one electrical cycle. It becomes therefore a matter of interest to see how the one form of computation can be turned into the other. A good deal depends naturally on the rate at which the speed variation rises and falls, but for a sufficiently close approximation it may be taken as a sine or cosine curve. Then the angular velocity ω may be written as equal to

$$a + b \cos c\theta.$$

Where θ is the angle the crank has moved through, a is the mean value of the angular velocity, and b is its maximum variation from the mean, so that the speed oscillates between $(a + b)$ and $(a - b)$. If c be unity this oscillation occurs once in a revolution, but if $c = 2$ then it occurs twice, and so on.

This may be written

$$\omega = \frac{d\theta}{dt} = a + b \cos c\theta$$

or

$$dt = \frac{d\theta}{a + b \cos c\theta}$$

integrate and

$$ct = \mathrm{A} + \frac{2}{\sqrt{a^2 - b^2}} \tan^{-1} \frac{\sqrt{a - b} \tan \frac{c\theta}{2}}{\sqrt{a + b}}$$

where A is some constant.

Put $\theta = 0$ when $t = 0$ and therefore $\mathrm{A} = 0$

so that

$$t = \frac{2}{c\sqrt{a^2 - b^2}} \tan^{-1} \frac{\sqrt{a - b} \tan \frac{c\theta}{2}}{\sqrt{a + b}} \quad . \quad . \quad . \quad (1)$$

Now $\frac{b}{a}$ is *half* the cyclic irregularity and may be given a symbol—call it m.

therefore
$$t=\frac{2}{ac\sqrt{1-m^2}}\tan^{-1}\frac{\sqrt{1-m}\,\tan\frac{c\theta}{2}}{\sqrt{1+m}}$$

$$\frac{\sqrt{1-m}\,\tan\frac{c\theta}{2}}{\sqrt{1+m}}=\tan\frac{act\sqrt{1-m^2}}{2}$$

or
$$\theta=\frac{2}{c}\tan^{-1}\left\{\frac{\sqrt{1+m}}{\sqrt{1-m}}\tan\frac{act\sqrt{1-m^2}}{2}\right\} \quad . \quad . \quad . \quad (2)$$

Now as m is always small compared to unity $\sqrt{1+m}$ may be written as $(1+\frac{1}{2}m)$ and m^2 be neglected;

therefore
$$\theta=\frac{2}{c}\tan^{-1}\left\{(1+m)\tan\frac{act}{2}\right\}.$$

Now were m really zero this equation would give to θ a value equal to

$$=\frac{2}{c}\tan^{-1}\tan\frac{act}{2}=at.$$

Call this value θ_0. Really it means the position of the crank at the instant determined by t if the angular velocity were strictly uniform and equal to a.

We may therefore write $\theta=\frac{2}{c}\tan^{-1}\left\{(1+m)\tan\frac{c\theta_0}{2}\right\}$ (3)

which is the solution.

If, for example, $c=1$ and $m=\frac{1}{200}$

then
$$\theta=2\tan^{-1}\left\{1{\cdot}005\tan\frac{\theta_0}{2}\right\}.$$

From this we see that when $\theta_0=120°$, θ becomes

$$\begin{aligned}&2\tan^{-1}\{1{\cdot}005\tan 60°\}\\&=2\tan^{-1}(1{\cdot}005\times 1{\cdot}73205)\\&=2\tan^{-1}1{\cdot}74071\\&=2\times 60{\cdot}12°\\&=120{\cdot}24°\end{aligned}$$

or that the crank would be nearly $\frac{1}{4}$ degree ahead of its supposed position.

It is useful to get an expression for this deviation directly. From (3)

$$\theta-\theta_0=\frac{2}{c}\tan^{-1}\left\{(1+m)\tan\frac{c\theta_0}{2}\right\}-\theta_0.$$

Find the maximum value of $\theta-\theta_0$ by differentiating and equating to zero. Then

$$(1+m)=\left\{1+(1+m)^2\tan^2\frac{c\theta_0}{2}\right\}\cos^2\frac{c\theta_0}{2}$$

$$=\cos^2\frac{c\theta_0}{2}+(1+m)^2.\sin^2\frac{c\theta_0}{2}$$

$$1=2\sin^2\frac{c\theta_0}{2}+m\sin^2\frac{c\theta_0}{2}$$

$$\sin^2\frac{c\theta_0}{2}=\frac{1}{2+m}$$

$$\tan^2\frac{c\theta_0}{2}=\frac{1}{1+m}$$

or $$\tan\frac{c\theta_0}{2}=1-\tfrac{1}{2}m \text{ approximately,}$$

so that the maximum value of $\theta-\theta_0$ is found from the expressions

$$\theta-\theta_0=\frac{2}{c}\tan^{-1}(1-\tfrac{1}{2}m)(1+m)-\theta_0$$

and

$$\theta_0=\frac{2}{c}\tan^{-1}(1-\tfrac{1}{2}m).$$

Therefore the maximum deviation

$$=\frac{2}{c}\left\{\tan^{-1}(1+\tfrac{1}{2}m)-\tan^{-1}(1-\tfrac{1}{2}m)\right\}$$

$$=\frac{2}{c}\tan^{-1}\frac{4m}{8-m^2}$$

If m be so small that m^2 can be neglected—as it practically is—this reduces to

$$\textbf{maximum deviation}=\frac{2}{c}\tan^{-1}\frac{m}{2} \quad . \quad . \quad . \quad (4)$$

If for example $c = 1$ and $m = \frac{1}{25}$, this equation gives the maximum deviation

$$= 2 \tan^{-1} \frac{1}{50} = 2 \times 1{\cdot}2 = 2{\cdot}4 \text{ degrees.}$$

When m is much smaller than $\frac{1}{25}$, say $\frac{1}{200}$ equation (4) can be approximately written :—

$\frac{2}{c} \times \frac{m}{2} = \frac{m}{c}$. So that with $c=1$ and $m = \frac{1}{200}$ the maximum deviation would be $\frac{1}{200}$ radian or 0·28 degree.

Equation (2) shows how the value of θ can be calculated for any position of the ideal crank, and the deviation may have its most important effect electrically even when it has not itself its largest numerical value. For that reason it is desirable to have some means of calculating it easily. In cases in which it is only desired to find the maximum deviation to some approximate degree of accuracy, it is sufficient to take a mean value of the excess angular velocity and multiply it by the time during which it operates. Thus if as before $c = 1$ and $m = \frac{1}{200}$, calling the angular velocity ω, the average excess of angular velocity

$$= \frac{2}{\pi} \times \frac{\omega}{200} = \frac{\omega}{100\pi}$$

and this operates through 180° or for a time equal to $\frac{\pi}{\omega}$, so that the angular motion gained

$$= \frac{\omega}{100\pi} \times \frac{\pi}{\omega} = \frac{1}{100}$$

and $\frac{1}{100}$ radian = 0·57 deg. This, divided between the two ends of the period, gives a maximum deviation of 0·28 deg., which agrees with the 0·28 deg. previously found.

For these values of c and m it may be said that the maximum deviation is about $\frac{1}{4}$ of a degree. If the alternator has six pairs of poles giving six electrical cycles during one mechanical one this deviation could also be called $\frac{1}{4} \times 6$ or one and a half electrical degrees, which corresponds with Mr. Schüler's result.

110. Balancing.—The problem of balancing the parts of a gas engine and providing for uniformity of torque as far as possible does not differ in principle from the corresponding problem in the case of the steam engine, and the author does not propose therefore to devote a great deal of space to this subject. The student should refer to what has been written on the subject of balancing by Professor Perry and Professor Dalby, both of whom have made a special study of the matter. It will, however, be advisable to give here a brief account of the general principles involved, leaving the application to be made to each and every problem as it presents itself. For it must be remembered that although the problem is often surrounded by complications which lead to the mathematical work looking difficult and involved, there is really no special difficulty about it at all, but merely a necessity that the fundamental principles should be rightly applied and that the algebraic or arithmetical work should be carried through without mistakes.

The simplest kind of balancing is that in which a flywheel is light on one side and requires a weight (W) fastened to the other side in order to prevent any jumping or vibration when the wheel rotates. This does not of necessity mean that an equal weight must be added to the other side, because it does not follow that it will be possible to place the balance weight at the same distance from the centre of the shaft, and the centrifugal force being equal to $\omega^2 r \dfrac{W}{g}$ (where ω=angular velocity in radians per second and r = distance in feet from the centre of the shaft) it is evident that the product of the W and the r in the balance weight must come out to a certain amount. If therefore the r is very small then W must be proportionately greater, and inasmuch as the balance weight is often bolted in between the spokes it is clear that r will usually be less than

the radius of the rim of the wheel. This is the simplest kind of balancing. The most complicated kind occurs in the motion of a rod, like a connecting-rod, in which one end reciprocates to and fro in a straight line and the other end follows a circular path, with the result that intermediate parts of the rod follow a complicated curve and one not easy to treat. In such a case as this it is customary to obtain an approximate solution by assuming that a certain part of the rod is massed at the cross-head and the rest at the crank-pin, and it is not unusual to make this division of the rod in inverse proportion to the distance of the centre of gravity from either end. This is only an approximation, unless it happens that the rod is so made (which it usually is not) that if hung up from the big and little ends in turn it will swing, pendulum wise, with the same number of swing-swangs per minute. When a number of rotating masses (real or assumed) have to be balanced it is useful, following Dalby's method, to consider the plane perpendicular to the shaft in which one or more of them lie to be rotating at the same speed as the shaft and to draw out on this plane the force diagram.

111. The connecting rod influences the problem of the running of the engine in another way. If the crank-pin rotated uniformly and the connecting rod were infinitely long the motion of the piston would be Simple Harmonic, and the displacement of the piston from the middle of its stroke would be $r \cos \theta$ at the instant when the angle between the crank and the line of dead centres was θ, the radius of the crank-pin circle being r. But the connecting rod in actual engines is usually quite short, never more than ten times the length of the crank arm, and usually much less. This produces a complicated motion of the piston, and it will be useful to calculate exactly what it is. Let P be the crank-pin and A the piston which, in the position shown, is at a distance AB from the beginning of its stroke. The angles θ and φ are as shown in the diagram. OP is r and AP, l. AB will be written as x. Now it is clear that

$$r \cos \theta + l \cos \varphi + x = \mathrm{BO}$$

and

$$\mathrm{BO} = l + r$$

so that

$$r \text{ ocs } \theta + l \cos \varphi + x = l + r \quad . \quad . \quad . \quad . \quad (1)$$

Also we have $\qquad r \sin \theta = l \sin \varphi \quad . \ . \ . \ . \ . \ . \ . \qquad (2)$

It is necessary to combine these two equations so as to find x in terms of known quantities.

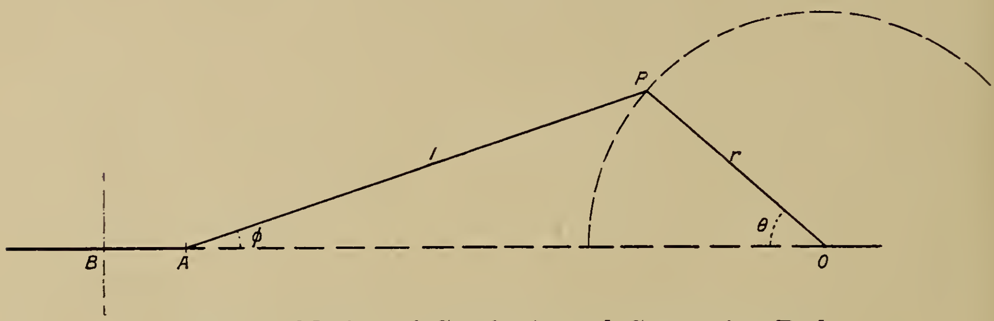


FIG. 54.—Motion of Crank-pin and Connecting Rod.

From (1)

$$x = l + r - r \cos \theta - l \cos \varphi$$
$$= l\,(1 - \cos \varphi) + r\,(1 - \cos \theta)$$

Also from (2)

$$\sin \varphi = \frac{r}{l} \sin \theta$$

and
$$\cos \varphi = \sqrt{1 - \frac{r^2}{l^2} \sin^2 \theta}$$

therefore $x = l\left(1 - \sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}\right) + r(1 - \cos \theta) \quad . \ . \ . \quad (3)$

and this gives the value of x for any value of θ.

Previously we have spoken of the distance of the piston from the mid point of its stroke rather than from either end, and it is useful to follow the same procedure here—call the displacement of the piston from mid stroke y—then $x + y = r$ or

$$y = r - x$$

so that
$$y = r \cos \theta - l\left(1 - \sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}\right)$$

furthermore θ is a function of the time, and since uniformity of rotation is assumed it will be directly proportional to the time. Put therefore $\theta = \omega t$

so that $y = r \cos \omega t - l\left(1 - \sqrt{1 - \frac{r^2}{l^2}\sin^2 \omega t}\right) \quad . \ . \ . \quad (4)$

Since, however $\left(\frac{r}{l}\right)^2$ is a small amount in all engines an approximation to the above may be written as

$$y = r\cos\omega t - l\left(1 - 1 + \tfrac{1}{2}\frac{r^2}{l^2}\sin^2\omega t\right)$$

$$= r\cos\omega t - \frac{r^2}{2l}\sin^2\omega t$$

or

$$y = r\cos\omega t - \frac{r^2}{4l}(1 - \cos 2\omega t) \quad . \quad . \quad . \quad (5)$$

This very interesting result shows that the position of **the piston can be stated as the sum of two S.H.M.**'s one of which corresponds to an infinitely long connecting rod and the other to a S.H.M. of twice the periodicity and of an amplitude depending on the ratio of r to l. The motion in fact is analogous to that of the air set into vibration by an organ pipe which in addition to giving its fundamental note gives also a weak first harmonic. Although this first harmonic is weak in its effect on the displacement of the piston, it is considerably more potent when velocities and accelerations have to be taken into account, as will presently appear.

Since from (5)

$$y = r\cos\omega t - \frac{r^2}{4l}(1 - \cos 2\omega t)$$

$$\frac{dy}{dt} = -\omega r\sin\omega t - \frac{r^2\omega}{2l}\sin 2\omega t$$

$$= -\omega r\left(\sin\omega t + \frac{r}{2l}\sin 2\omega t\right) \quad . \quad . \quad . \quad (6)$$

and

$$\frac{d^2y}{dt^2} = -\omega^2 r\left(\cos\omega t + \frac{r}{l}\cos 2\omega t\right) \quad . \quad . \quad . \quad (7)$$

It is important to note that expression (6) which gives the velocity of the piston at any point has the multiplier $\frac{r}{2l}$ in front of the harmonic term, and that expression (7) which gives the acceleration and therefore measures all the inertia forces has the multiplier $\frac{r}{l}$. It follows therefore that the

three multipliers in the harmonic term for displacement, velocity and acceleration run thus, $\frac{r}{4l}$, $\frac{r}{2l}$ and $\frac{r}{l}$, showing that a ratio of $\frac{r}{l}$ which will produce a 5 per cent. difference in the position of the piston will bring about a 10 per cent. change in the velocity and about 20 per cent. in the acceleration. It will now be realized that, when forces are being nicely balanced, the importance of the harmonic term must be carefully allowed for.

It is often useful to bear in mind a simple rule for the value of the acceleration at the ends of the stroke, i.e. when $\omega t = 0°$ or 180°. From formula (7) it will be seen that this leads to

$$\frac{d^2y}{dt^2} \text{ either} = -\omega^2 r\left(1+\frac{r}{l}\right), \text{ or} = -\omega^2 r\left(-1+\frac{r}{l}\right)$$

$$= \mp\omega^2 r\left(1 \pm \frac{r}{l}\right)$$

a very simple rule, i.e. that the acceleration at the end of the stroke is more or less than the S.H.M. value by the fraction $\frac{r}{l}$ of that value.

112. Connecting Rod Effect.—This is best illustrated by a geometrical construction due to Professor J. Harrison.

The construction is as follows :—

OB is the crank and AB the connecting rod of which G is the centre of gravity.

OQ and SH are perpendicular to AO.

Ha is perpendicular to AB.

SQ and Gg are parallel to AO.

GU $= k^2/$AG where $k =$ radius of gyration.

UX is parallel to Ba.

Then TXN parallel to gO is the line of action of the resultant of all the forces acting on the rod and its value in mq^2.XN where $m =$ mass of rod and $q =$ angular velocity of crank-pin. The proof of this construction is given in Perry's *Steam Engine*, and may there be referred to by those interested.

This diagram enables the direction and amount of the inertia

forces due to the connecting rod to be calculated for each position of the crank.

If k^2 happens to be equal to the product of AG and GB, then $GU = k^2/AG = GB$ so that the point U would coincide with B and the resultant force would pass through O and hence there would be no " whipping effect " of the rod. One often

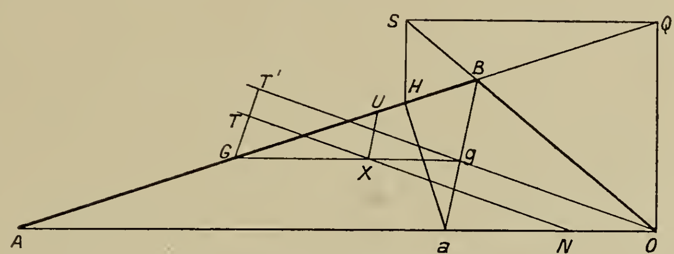


FIG. 55.—Resultant of all the Accelerating Forces on a Connecting Rod.

sees connecting rods produced beyond the crank-pin with the object of bringing about this relationship. When it is accurately obtained it will be found that the period of swing of the rod about either big or little end will be the same.

EXAMPLES

1. In the case of a single-acting gas engine working on a 4-stroke cycle, the mean effective pressure is 40 lb. per sq. inch, the diameter of cylinder 8 in., the stroke 8 in., the number of revolutions 360 per minute. The governor cuts out 1 explosion every 24 revs. Calculate the I.H.P. developed by the engine.

2. Find the I.H.P. of a gas engine of which the piston is 12 in. diameter, its crank is 8 in. long, the engine makes 160 revolutions or 80 cycles per minute, and 30 per cent. of the possible explosions are omitted. The mean area of all the diagrams on a card taken with a 120 spring in the indicator as measured by the planimeter is 2·62 sq. inch ; length of diagram parallel to atmospheric line 4·03 in.
(B. of E., 1899.)

3. The mean effective pressure on the piston, both in the forward and back strokes, is 62 lb. per sq. inch ; cylinder 18 in. diameter ; crank 18 in. long. What is the work done in one revolution ?
(B. of E., 1906.)

4. The following data arose in the trials of a gas engine :—Stroke 23 in. ; diameter of piston $16\frac{1}{2}$ in. ; average M.E.P. 68·8 lb. per sq.

inch ; number of explosions per min. 95 ; circumference of brake-wheel 24·5 ft. ; average *net* load noted in brake test 484·6 lb. ; average speed in r.p.m. 190.

Calculate (i) The I.H.P.
(ii) The B.H.P.
(iii) The mechanical efficiency per cent.

5. The following data are taken from a record of a test of a gas engine using power-gas :—cylinder diameter = 48 in., stroke = 54 in., M.E.P. = 75 lb. per sq. inch. Number of explosions per min. = 36. Gas used per min. = 1020 cu. ft. Calorific value of gas = 60 C.H.U. per cu. ft. B.H.P. = 545.

Calculate (i) The I.H.P.
(ii) The mechanical efficiency of engine.
(iii) Volume of gas used per I.H.P. hour.
(iv) Volume of gas used per B.H.P. hour.
(v) Indicated thermal efficiency.
(vi) Brake-thermal efficiency.

6. The M.E.P. in the cylinder of a gas engine is 92 lb. per sq. inch when the speed is 166 revs. per min. and there are 72 explosions per min. At the same time, the torque exerted by the crank-shaft is determined by a dynamometer to be 1,440 lb. ft.

Calculate (i) I.H.P.
(ii) B.H.P.
(iii) Mechanical efficiency.

The cylinder is 14 in. in diameter with 22 in. stroke.

(B. of E., 1912.)

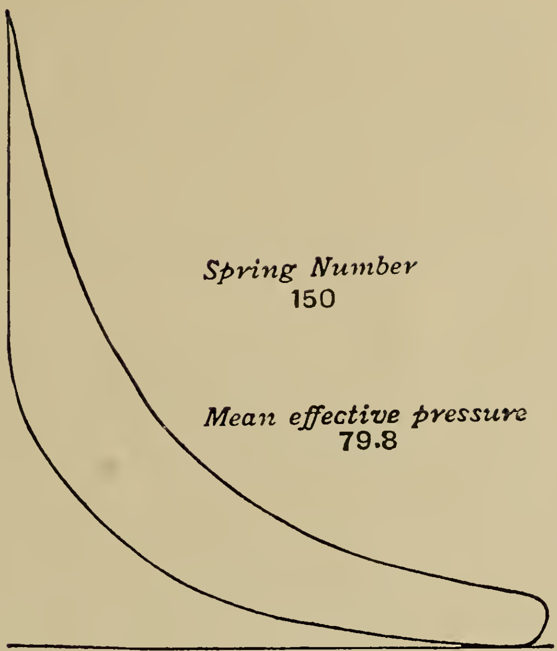
7. When a gas engine is running fully loaded the temperature of the exhaust gases left in the clearance space at the end of the exhaust stroke is 700° C., and the temperature of the gas and air sucked in just before they enter the cylinder is 100° C. The clearance space is a quarter of the total cylinder volume (including clearance space). Show that the temperature of the gases filling the cylinder at the end of the suction stroke will be 170° C. Assume that no heat is lost to or gained from the cylinder walls during suction, that the pressure inside the cylinder is the same as that of the atmosphere, and that the specific heat of the exhaust gases and of the incoming charge is the same constant quantity. (Mech. Sc. Tripos, 1904.)

8. A gas engine working on the Otto cycle, and running at 200 revs. per min., has a cylinder $11\frac{1}{2}$ in. diameter, and stroke 21 in., and the compression space is 0·185 of the stroke volume. At the end of the suction stroke the cylinder is filled with gas and air at a pressure of 14·7 lb. per sq. inch, and a temperature of 100° C. The cylinder contents consist of 1 of gas to 10 of air. The calorific

value of the gas is 320 C.H.U. per standard cubic foot.

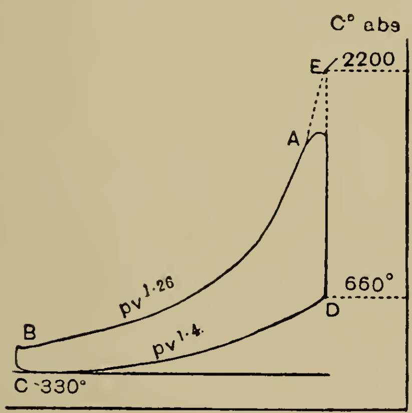
The accompanying indicator diagram is taken from the engine: find the maximum power and the thermal efficiency of the engine.

State how you would determine the heat given to the cylinder walls during compression and explosion.



9. The indicator diagram of a gas engine is shown in below. The equation to the expansion curve AB is $pv^{1\cdot 26}$ = constant, and to the compression curve CD is $pv^{1\cdot 4}$ = constant. The curve AB if produced meets the vertical explosion line in E. The absolute temperatures of E, D, and C are indicated on the diagram. Determine the thermal efficiency of the cycle, neglecting the rounded corners of the diagram, and taking the working substance as air.

(Mech. Sc. Tripos, 1912.)



10. Crank 1 ft., connecting rod 4·5 ft.; what are the accelerations at the ends and some other point in the stroke, if the engine makes 200 revolutions per minute? The piston and rod and crosshead are 420 lb.; draw a diagram to show the force in pounds required to produce the motion. State the scale clearly.

(B. of E., 1906.)

11. A piston and rod and crosshead weigh 330 lb. At a certain instant, when the resultant total force due to steam pressure is 3 tons, the piston has an acceleration of 370 ft. per second in the same direction. What is the actual force acting at the crosshead?

(B. of E., 1902.)

12. In a gas engine release occurs at seven-eighths of the stroke and at a pressure of 40 lb. per sq. inch absolute. The clearance space is a quarter of the total cylinder volume. The engine works on the Otto

cycle, explodes every time and is not scavenged. The mixed gas and air just before being drawn into the cylinder on the suction stroke has a temperature of 100° C. Estimate the temperature of the charge filling the cylinder at the end of the suction stroke.

In making your estimate you will probably assume that gases before and after explosion behave as the same perfect gas. How far is this assumption correct? Illustrate the possible errors in estimates of temperature based on this assumption by finding their amount in the case of a mixture of one volume of hydrogen and five of air.

(Mech. Sc. Tripos, 1904.)

13. A flywheel 4 ft. diameter in the form of a disc 6 in. thick is keyed to the shaft of a high-speed engine. Calculate how much energy it stores at 500 r.p.m. and find how much the store of energy is increased when the speed increases 5 per cent. One cu. ft. of material weighs 480 lb. (B. of E., 1912.)

14. What must be the size of a flywheel in order that the maximum speed may not exceed the mean speed of 60 revs. per min. by more than 0·2 rev. per min. when the area of the crank-effort curve cut off by the mean crank-effort line represents 12·5 ft. tons.

Give the mean radius of the flywheel and the weight in tons of the rim and work out the dimensions on the assumption that the mass of the wheel is all concentrated at the mean radius of the wheel and that the speed at the mean radius is limited to 50 f.p.s. (B. of E., 1912.)

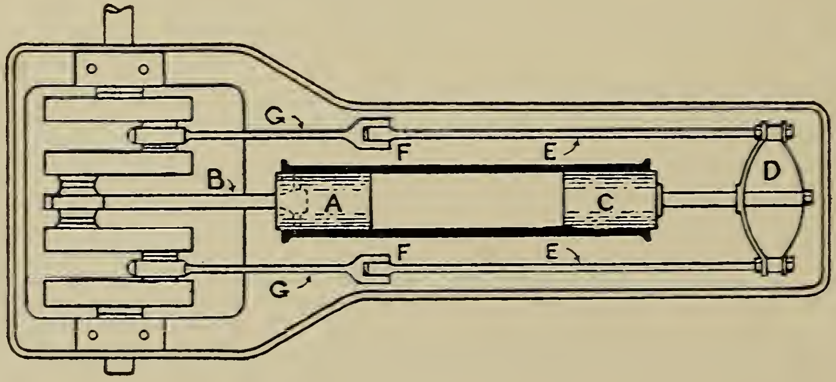
15. Plot roughly the acceleration of the piston masses in a reciprocating engine from the following data:—

Stroke 2 ft.
Connecting rod 4 ft.
Speed 200 revs. per min.

Write down the accelerations at the beginning and end of stroke.

(B. of E., 1912.)

16. The Figure shows diagrammatically the moving parts of an Oechelhauser gas engine. The piston A is coupled direct to the central crank-pin by connecting rod B; the piston C is coupled to the outer crank-pins through a crosshead D, side-rods E, and connecting rods F. The pistons move in opposite directions, and the explosion takes

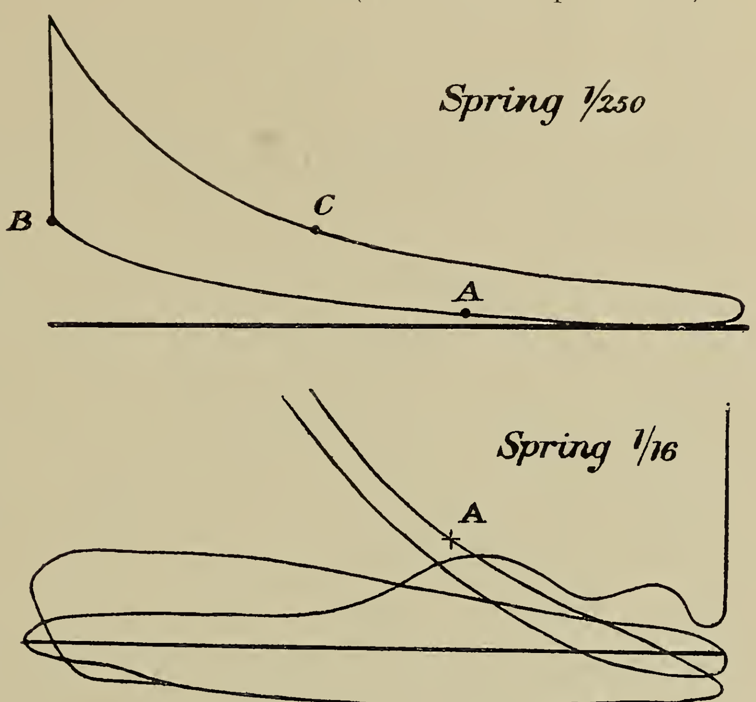


place between them. The mass of piston C with its attached reciprocating parts is 8 tons, that of A is 4 tons. The stroke of each piston is 43 in., and length of each connecting rod is 108 in. Find the magnitude and direction of the unbalanced horizontal inertia force on the engine at each dead-point, when the speed is 100 revs. per min.

It is often assumed in the calculation of inertia forces that the connecting rod may be treated as though half its mass were concentrated at each end. Discuss this assumption and explain its practical consequences in regard to the problem of balancing.

(Mech. Sc. Tripos, 1913.)

17. The two indicator diagrams * herewith were taken from a gas engine. The temperature at the point A was measured directly and found to be 100° C. The corresponding pressure and volume can be found from the diagram. Calculate the temperature at the point B. What further data are required to enable you to calculate the temperature at a point C on the expansion line ?



The pressure of the atmosphere is 14·7 lb. per sq. inch, and the clearance is $\frac{1}{5}$ of displacement volume.

The cylinder of the engine is 5·5 in. diameter and the stroke 10 in. The B.H.P. of the engine was varied, and the speed was maintained (by a hit-and-miss governor) approximately constant at 370 r.p.m. during the trials. Determine from the diagrams and the following data, the frictional H.P. of the engine for each loading :—

B.H.P.	5·51	3·78	1·6	0
Explosions per min. . .	156	120	75	44

(B. of E., 1913.)

* These two diagrams are not really consistent, but the question is one worth study.

CHAPTER VI

The Gas Producer

Theory—Typical Suction and Pressure Producers—Tests—Costs—Use of Gas Producer for Marine Purposes—Appendix containing Description of Mode of Operation of Suction Gas Plant.

113. Producer Gas. *Theory.*—In a steam boiler the energy stored up in the coal is liberated by combustion in an atmosphere containing oxygen. In other words, heat is liberated by the combination of the carbon with oxygen first to form CO, and then, if enough air be present to add a further atom of oxygen to the molecule, to CO_2. When 12 kg. of carbon (that is to say the atomic weight of carbon taken in kilograms) are oxidized to CO, 29,400 calories * are given off, and when CO_2 is formed a further 68,200 calories are liberated, making a total of 97,600. This means that if the carbon be only oxidized to the CO stage not more than about 30 per cent. of the available heat energy is given up, and that by far the most of the available heat is obtained from the stage in which CO becomes CO_2. Even supposing that in a given steam boiler the whole of the 97,600 calories were given off from each 12 kg. of carbon (neglecting for the moment the hydrogen and hydrocarbons in the coal) only a fraction, not greater than 70 per cent., usually gets to the water, and the balance goes away up the chimney or is lost by radiation. With gas producers such heavy losses do not occur. Their efficiency depends upon the working process, but it may be taken as being seldom less than 80 per cent. and often as much as 90 per cent. even when working with **anthracite coal** and

* In this chapter the calorie is the kilogram-calorie.

not chemically pure carbon. In a gas producer, air is forced or drawn through a mass of highly heated fuel, with the result that the carbon is oxidized. Also, in order to keep the temperature within reasonable limits, and for another reason to be given later, steam is admitted along with the air and both together pass upwards through the glowing fuel.

When the air and steam are forced through by pressure the producer is called a **Pressure Producer.** When however they are drawn through by suction caused by the suction strokes of the engine, they are known as **Suction Producers.** The theory in both cases is similar.

It may seem strange to those who approach the subject for the first time that it should be possible for the gas given off to contain as much as 80 to 90 per cent. of the total heat energy in the coal. It would be apparent to them from their knowledge of chemistry that even if pure CO came away from the producer, and no CO_2 at all, there would be a loss of the 30 per cent. of energy liberated when the carbon was oxidized to CO, leading to an **apparent maximum efficiency of 70 per cent.** The explanation is that this 30 per cent. is not lost. It serves to keep the furnace alight, and to decompose the entering steam into hydrogen and oxygen, thus—

$$2\,H_2O = 2\,H_2 + O_2 \quad . \quad . \quad . \qquad (1)$$

and in so doing it stores up 116,400 calories for each 36 kg. of steam decomposed. It is easy to see that by suitably balancing the proportions of air and steam admitted, it is possible to absorb the greater part of the 30 per cent. of energy liberated by the formation of CO, and to carry it as potential chemical energy to the gas engine, where the hydrogen and oxygen can again unite. In reality it is not quite so simple as this, because the oxygen from the decomposed steam has also to pass over glowing carbon, with the result that a further supply of CO is formed. Radiation of heat from the producer prevents the efficiency being 100 per cent.

Following generally the procedure adopted by Mr. Dowson, who invented the first of these plants, the reactions may be semi-mathematically stated thus :—

Taking weights equal to molecular weights in kg.

Carbon-monoxide is thus formed :—

$$2C+O_2=2CO+58,800 \text{ calories} \quad . \quad . \quad . \quad . \quad (2)$$

Carbon-dioxide would be formed thus :—

$$C+O_2=CO_2+97,600 \text{ calories} \quad . \quad . \quad . \quad (3)$$

The former of these two equations gives a gas having a calorific value of about 119 B.Th.U. per cubic foot, but when **steam** is admitted this value rises rapidly owing to the hydrogen present.

As already stated the decomposition of steam follows

$$2H_2O=2H_2+O_2-116,400 \text{ calories} \quad . \quad . \quad . \quad (4)$$

the negative sign meaning that heat is absorbed and not liberated. The oxygen so produced also joins in the reaction, so that one of the following formulæ

$$H_2O+C=H_2+CO-28,800 \text{ calories} \quad . \quad . \quad . \quad (5)$$

$$\text{or } 2H_2O+C=2H_2+CO_2-18,800 \text{ calories} \quad . \quad . \quad (6)$$

is followed in the decomposition of the steam. In both (5) and (6) an absorption of heat takes place which allows of a balance being obtained by a careful regulation of the relative proportions of air and steam admitted.

114. It is useful to discover **what quantity of water is theoretically required** per pound of coal in order to keep this reaction balanced.

Assume that the reaction follows equations (2) and (6). Really it will not follow quite such simple laws, but it will approximate thereto if the temperature is high enough.* Equation (2) shows that for each 24 kg. of carbon used 58,800 calories will be liberated, and equation (6) that 18,800 calories will be absorbed by each 36 kg. of steam dissociated, requiring also for its dissociation 12 kg. of carbon. To absorb the whole of the 58,800 calories liberated $36\times\frac{58,800}{18,800}$ kg. of steam would be required. But the steam is not admitted to the producer as steam, but as water, and there is therefore the

* According to Robson, equation (5) is followed at a temperature of about 600° C. ; and equation (6) from 900 to 1000° C. At temperasures between 600 and 1000 both reactions occur. The equilibrium state is a function of both temperature and time.

latent heat of evaporation to be considered. Now the latent heat of 36 kg. of water-vapour at 20° C. is 21,600 calories, and this must be added to the 18,800 calories due to chemical dissociation, making a total of 40,400 calories, so that only $36 \times \frac{58,800}{40,400}$ kg. of water would really be required, and this works out at 52·4 kg. of water. The quantity of carbon corresponding to this is clearly $24 + \left(\frac{12}{36} \times 52{\cdot}4\right) = 24 + 17{\cdot}5 = 41{\cdot}5$ kg. of carbon. So that $\frac{52{\cdot}4}{41{\cdot}5}$ or 1·26 kg. of water *will be required for each kg. of carbon.*

The next point to determine is the nature of the mixture of gases given off in this way. Equation (2) shows that for each 24 kg. of carbon there will be given * off $22{\cdot}4 \times 2 \times 1,000$ litres = 44,800 litres of CO Equation (6) adds to this an equal volume of hydrogen and half the volume of CO_2 for each 12 kg. of carbon. Now the quantities in equation (6) must clearly be proportional to 17·5 and not 12 kg. of carbon, and therefore the volume of hydrogen will be $\frac{17{\cdot}5}{12} \times 44,800 = 65,200$ litres and the volume of CO_2 will be 32,600 litres. The total will therefore be

CO :—	44,800 litres
CO_2 :—	32,600 ,,
H_2 :—	65,200 ,,
	142,600 litres or 142·6 cubic metres.

But it must be remembered that in equation (2) oxygen is supplied to the extent of 22,400 litres, and that as this is drawn from the air it will be accompanied by $\frac{79}{21} \times 22,400$ litres of nitrogen which will pass through without change. So that to the above table must be added $\frac{79}{21} \times 22,400 =$

* Based on the principle that the molecular weight of any gas taken in grams will occupy a volume of 22·4 litres. (Some recent work has been based on a revised figure of 22·25 litres.)

84,100 litres of nitrogen, making the total and proportions thus :—

CO . .	44,800	litres	or 19·8 per cent.
CO_2 . .	32,600	,,	or 14·4 per cent.
H_2 . .	65,200	,,	or 28·8 per cent.
N_2 . . .	84,100	,,	or 37·0 per cent.
	226,700		100·0

Thus 226,700 litres of gas are given off for each 41·5 kg. of carbon or 5,450 litres per kg. of carbon, and 5,450 litres is of course 5·45 cubic metres.

What is the **calorific power of this producer gas**? The N_2 and CO_2 can give nothing. The CO will yield (97,600 — 29,400) calories for each 28 kg. of CO, or $\frac{68,200}{28} = 2,440$ calories per kg. of CO. The H_2 will yield 116,400 calories per 4 kg. of gas, or 29,100 calories per kg. of hydrogen. Take 1 cubic metre or 1,000 litres of the producer gas. It will contain 198 litres of CO yielding $\frac{198}{22,400} \times 68,200 = 602$ calories, and of hydrogen $\frac{288}{22,400} \times 58,200 = 750$ calories, making a total of 1,352 calories per cubic metre. Furthermore the steam formed by the union of the hydrogen and oxygen will be capable of yielding up its latent heat, which will add 21,600 calories for each 4 kg. of hydrogen concerned. Now the weight of the hydrogen in 1,000 litres of the gas is $\frac{288}{22,400} \times 2$ kg. and the calories in the latent heat of the steam will therefore be $\frac{288}{22,400} \times 2 \times \frac{21,600}{4} = 139$ calories, which when added to the 1,352 calories found above, makes a total calorific value of 1,491 calories per cubic metre of the gas given off by the producer. In cases in which the latent heat of the steam formed cannot be utilized, it is customary to use the lesser value of the calorific constant, and write it down in this case as 1,352 calories only, which is nearly 10 per cent. less. The

figure of 1,491 calories per cubic metre corresponds to 168 B.Th.U. per cubic foot.

115. Dowson has carried out calculations similar to the above for a number of possible reactions, and the following tables show some of the results he has found.

Reaction between Air and Carbon: proportions of CO and CO_2 formed per cent. by volume, depending upon the temperature of the reaction		Composition of gas per cent. by volume (Steam decomposed according to equation (6))				Steam used per kilo of Carbon	Gas formed per kilo of Carbon	Calorific Power of Gas made	
CO	CO_2	CO_2	CO	H_2	N_2	Kilos.	Cubic Metres	Calories per Cubic Metre	B.Th.U. per Cubic Foot
0	100	28·45	—	40·25	31·3	2·12	6·54	1,243	139·7
10	90	27·8	0·9	39·7	31·6	2·08	6·48	1,254	140·9
20	80	27·1	1·9	39·15	36·85	2·02	6·41	1,267	142·4
30	70	26·3	3·0	38·5	32·2	1·97	6·34	1,282	144·0
40	60	25·35	4·3	37·7	32·65	1·19	6·26	1,298	145·8
50	50	24·3	5·85	36·8	33·05	1·83	6·17	1,316	147·9
60	40	23·0	7·65	35·8	33·55	1·75	6·07	1,340	150·5
70	30	21·5	9·8	34·55	34·15	1·66	5·95	1,366	153·5
80	20	19·6	12·4	33·0	35·0	1·55	5·81	1,398	157·1
90	10	17·3	15·65	31·1	35·95	1·42	5·65	1,438	161·6
100	0	14·4	19·7	28·8	37·1	1·26	5·45	1,490	167·5

This table serves to show the very complete way in which Dowson worked out the chemical problems relating to producer gas, and the student who wishes to pursue such matters further is referred to that writer's interesting book on the subject.

We have now discussed the ideal conditions of working. In practice, about the theoretical weight of water is used in suction producers. For pressure producers such as the Mond producers an excess of steam is admitted in order that the temperature of the coal may be kept to a point lower than

that at which **ammonia** dissociates, it being a feature of this process to recover and sell the ammonia produced from the nitrogen contained in bituminous coals; the effect of this, incidentally, is to lower the thermal efficiency of the producer to about 80 per cent.

Equation (5) is sometimes followed instead of equation (6) for the decomposition of the steam, depending on the temperature of the reaction and the masses involved.* Mr. Dowson gives these two comparisons of the theory and practice in each case:—

Theory.		Practice.	
Gas formed according to Equations (2) and (5):—		Gas made at Millwall. 121·3 vols. contain same weight of carbon and consist of:—	
	Per cent. by Volumes.		Volumes.
CO	39·9	CO	33·5
H_2	17·0	H_2	18·6
N_2	43·1	N_2	62·8
		CO_2	4·7
		Methane	1·7
	100·0		121·3
Gas formed according to Equations (2) and (6):—		Gas made at Winnington. 117·6 vols. contain same weight of carbon and consist of:—	
	Per cent. by Volumes.		Volumes.
CO	19·7	CO	12·9
H_2	28·8	H_2	34·1
CO_2	14·4	CO_2	18·8
N_2	37·1	N_2	49·4
		Methane	2·4
	100·0		117·6
		It will be noticed that an excess of air has been admitted in each case.	

116. Actual Producers. In Fig. 56 is shown a reproduction of a working drawing of a 150 H.P. suction producer made by the Campbell Gas Engine Co. The steam required for the reaction is derived from the annular boiler surrounding the gas producer, and the heat necessary for vaporization is derived

* *See* footnote on p. 172.

from the heat of the fuel. This steam passes with the air down a pipe leading to the base of the gas producer, and is then drawn through the glowing fuel which is maintained at a temperature of about 1,000° C. The air and steam on passing through the furnace are decomposed in accordance with the equations already given, and the hot producer gas then passes through a dust trap or separator, and then past a water seal into the coke scrubber which consists of a tall vertical vessel containing coke upon which a water spray is kept playing. This cools the gas, condenses any steam there may be in it and serves

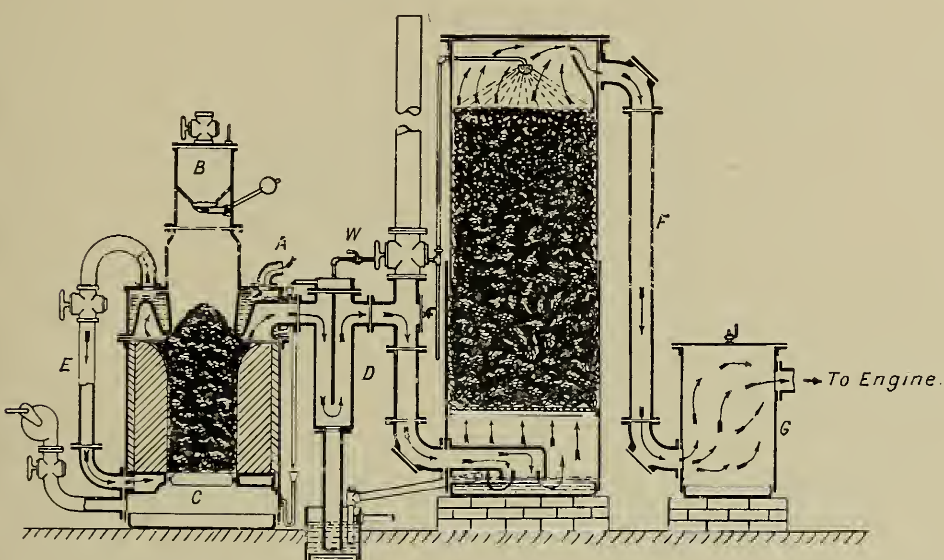


FIG. 56.—Sectional elevation of a 150 H.P. Campbell Suction Gas Producer, Fuel is first admitted through the hopper *B*. Air then enters at *A*, picks up steam on its way and passes by the pipe *E* to the grate *C*. The gases come away from the upper part of the producer and pass by the pipe system shown at *D* to the Scrubber Chamber, where they are cleansed and cooled. The gases are next drawn along the pipe *F* to the expansion box *G* on their way to the engine.

generally to cleanse it. Thence the gas passes to a gas box * to equalize the pressure, and from that it is drawn into the engine as wanted. A full description of how to work such a producer is, on account of its possible value to such readers who may be unacquainted with the actual working of such plant, given as an appendix to this chapter. The above

* This box should be put as near the engine as possible.

description applies to a plant using anthracite.* When it is desired to use coke as fuel, a sawdust scrubber is usually required in addition to the coke scrubber. An outside view of a similar plant is also given in Fig. 57.

There is not a great deal of difference between the different makes of suction producer plant. Fig. 58 shows an outside view of a National Gas Engine Co. type, similar to that which was awarded the gold medal at the Royal Agricultural Society's Trials in 1906. Its internal arrangements are much the same as those already described, except that the vaporizer

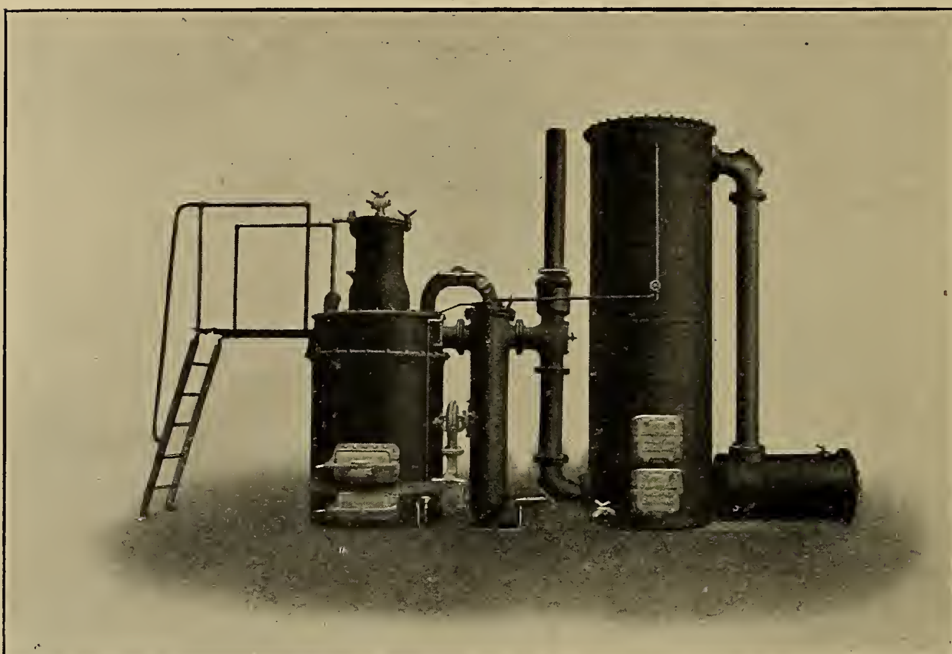


FIG. 57.—Outside view of 80 B.H.P. Campbell Suction Gas Plant. Note small size of Producer for the amount of power produced.

is fed with water which has first been heated by being passed through a pipe in the gas outflow passage and is then vaporized on the "flash" system.

Pressure producers are worked on much the same general principles, except that the air and steam are forced through the coal instead of being sucked through. In general, too, they are for much larger plants. Suction producers are usually

* Bituminous fuels cannot be used in suction gas producers, unless of the specially designed type made by Dowson and a few other makers, and such producers are less simple to operate.

for quite small outputs—commonly about 30 or 40 H.P. and rarely going beyond 500 H.P., whereas the power from pressure producers may run into thousands of horse-power,

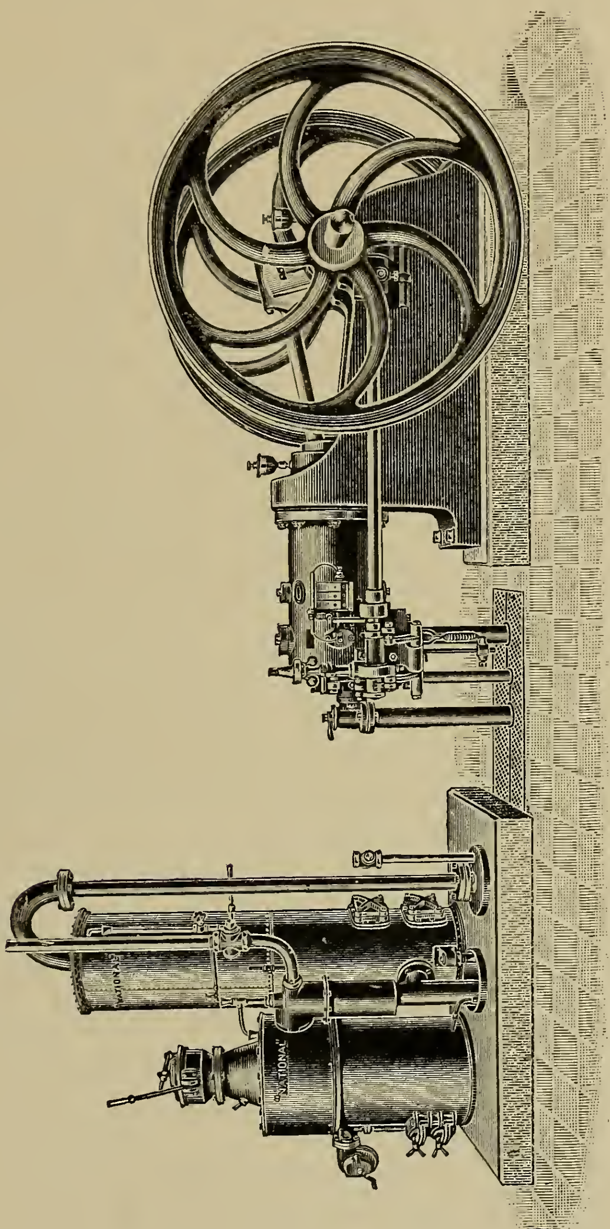


FIG. 58.—National Gas Engine and Suction Plant.

and the latter are therefore of a much more extensive nature, and a good deal more complicated, especially when a feature is made of by-product recovery.

117. Tests. It will be of interest to give here some figures

from the tests held on suction producer plant in 1905 by the Highland and Agricultural Society of Scotland, and in 1906 by the Royal Agricultural Society.

In the 1905 trials ten complete plants, exhibited by six different firms, were sent in for the competition. Particulars of these plants are given in the following table (*see* p. 181).

The result of the trials was given in the Judges' report,* of which the following contains an account.

Each plant was allowed half an hour of steady working before the actual power test, at the end of which the plant was brought back as nearly as possible to the same condition in respect of fuel, etc., as it was at the beginning of the trial, and the actual weight of fuel supplied in the interval was taken as that consumed by the plant during the power test. The obviously weak point in this procedure was that it was quite impossible to determine absolutely whether the plant was really in the same condition at the end of the trial as it was at the beginning. By running the test for a long enough time, however, any slight error in this respect could be rendered of little importance, and probably the method adopted was the best one. The alternative would have been to start the producers up from rest, and note the fuel put in, then at the end of the trial, note the proportion in the producer which had not been burnt, subtract the two, and add to this any fuel which had been introduced during the test. This procedure was adopted at the R.A.S. trials in 1906, except that the fuel consumed when the producers were banked up all night was also included, so leading to the disadvantage that it did not give a real fuel economy test. Also it was extremely difficult to tell at the end of the trial how much of the fuel left in the producer could properly be said to be "unburnt."

In the Scotch trials it was found that the coal per B.H.P.-hour at full load varied from 1·25 to 0·84 lb., and at half load from 1·55 to 0·91 lb. This was for the 8 H.P. sizes. For the larger, 20 H.P., plants the fuel per B.H.P.-hour at full load varied from 0·93 to 0·77 lb. and at half load from 1·08 to 0·92 lb. These results serve to show how economical the suction producer plant is when compared with steam engine

* *Engineering*, November 17, 1905.

General Dimensions and Particulars of Suction-Gas Plants Entered for Trial

—	Plants of about 20 Brake Horse-Power Capacity						Plants of about 8 Brake Horse-Power Capacity			
Exhibitor	The Aeme Engine Company, Shettleston, Glasgow	The Campbell Gas-Engine Company, Halifax	Messrs. Crossley Bros., Openshaw, Man-ehester	The Industrial Engineering Company, Hyde, near Manchester	The National Gas-Engine Company, Ashton-under-Lyne	Messrs. Tangyes, Ltd., Birmingham	The Campbell Gas-Engine Company, Halifax	The Industrial Engineering Company, Hyde, near Manchester	The National Gas-Engine Company, Ashton-under-Lyne	Messrs. Tangyes, Ltd., Birmingham
Declared capacity of gas-producer plant, B.H.P.	25	18	24	25	20	21	8	10	10	12
Price of gas-producer plant (complete) .	£94 10*s.*	£105	£80	£72	£80	£90	£80	£60	£65	£58
Total weight of plant, cwt.	35	25	45	25	25	28	25	15	20	17
General description of engine	"Acme"	Campbell	Crossley	"Acme"	"National"	Tangye	Campbell	"Acme"	"National"	Tangye
Declared brake horse-power of engine . .	22	18	16	22	20	19	8	13½	8	7½
Diameter of cylinder, in.	10	9½	8½	10	10	10	7	8½	7	7
Stroke . . . . . ,,	17	18	20	17	18	19	12	14	15	16
Revolutions per minute (declared)	220	200	200	220	190	190	230	230	220	220
Price of engine (complete)	£130	£125	£110	£130	£120	£131	£80	£75	£80	£90
Weight of engine, cwt.	60	66	54	60	78	69	29	25	40	40
Price of producer plant and engine (complete)	£224 10*s.*	£230	£190	£202	£200	£221	£160	£135	£145	£148
Space taken up by complete plant—producer, engine, etc. . . sq. ft.	144	225	240	288	225	225	160	288	225	160

plant of the same output: the latter would consume anything from 2½ times to 4 times as much fuel per B.H.P. Other interesting figures reported by the Judges are that the capacity of the producer per declared B.H.P. varied from 0·124 cu. ft. to 0·295 cu. ft. for the 20 H.P. size, and from 0·161 cu. ft. to 0·372 cu. ft. for the 8 H.P. size. Each of these figures show a ratio of about 2·3 to 1 and the price of the plants varied also but in not so great a ratio. The variation in cubic feet capacity per B.H.P. was an indication that little had then been done towards standardization of design.

118. The tests carried out by the R.A.S. in 1906 were considerably more elaborate, and, as already stated, a different procedure was followed. The report of the Judges had been published and, although in some aspects it may be said to be controversial, it is certainly worth study. Fourteen plants were entered for trial and all but three ran through to the finish. The capacity in each case was 15 to 20 H.P.* A list of the plants with their leading dimensions and other particulars is given here—

Name of Producer.	Name of Engine.	Revs. min.	Stroke In.	Diam. of Cyl. In.	Declared B.H.P. on Anthracite.
National	National	190	18	10	20
Dowson	Railway and General	170	18	12	20
Paxman	Paxman	220	15	9½	15·5
Dowson	National	190	18	10	20
Campbell	Campbell	200	19	9½	18
Campbell	Campbell	190	20	10	20
Dudbridge	Dudbridge	200	17	9¾	20
Mersey	Gardner	200	18	9	20
Hindley	Hindley	600	7	7	16
Kynoch	Kynoch	240	18	9	17
Newton	Newton	200	18	9	20
Fielding	Fielding	220	18	9½	18
Crossley	Crossley	220	21	8½	17
Crossley	Crossley	180	21	8½	15

Measurements made of the fuel and water consumption showed figures ranging from 1·47 to 1·04 lb. of anthracite

* Grate areas averaged about 8½ sq. ins. per B.H.P.

per B.H.P.-hour and from 3·61 to 0·73 gallons of water per B.H.P.-hour. The enormous variation in the quantity of water required was very striking, and it showed that there was a considerable difference in the manner of operation of the various plants. As the water required for steam making is very small, practically the whole of the above difference must have been due to the different quantities taken by the scrubber.

The Judges published the following conclusions as a result of the consumption trials—

That with a good suction producer plant, working continuously, at the specified loads and under the best conditions, the following results may be anticipated :—

With Anthracite.

Full load : 1·1 lb. per B.H.P.-hour including fuel needed for starting, and for banking during the night.
Half load : 1·6 lb. per B.H.P-hour including as above.
Water : 1 gallon per B.H.P.-hour at full load and ¾ gallon at half load.

With Coke.

Full load : 1·3 lb. per B.H.P.-hour including fuel needed for starting.
Water : 1½ gallons per B.H.P.-hour at full load.

Professor Dalby * also recorded as a result of these trials that—

"Assuming a 20 B.H.P. plant to start on Monday morning with an empty producer, and to run ten hours per day on full load for a week, banking the fires at night, the consumption of anthracite peas would be about half a ton for the week, and about ⅜ ton if the average load is about half full load. With coke the consumption is about 25 per cent. more. From 2,000 to 3,000 gallons of water per week are required for a 20 B.H.P. plant to provide water for the scrubber and the producer, and of this by far the larger part would be used in the scrubber."

Tests were also made of the times taken to start up and

* B. A. paper, August, 1906.

to change load. As a result of their investigations the Judges awarded the premier places to the National and Crossley plants. The Judges were Professor Dalby and Capt. Sankey, R.E.

119. Test of a Dowson Suction Gas Producer Plant.—The following account of tests on two Dowson Suction Plants is extracted from Mr. Dugald Clerk's 1904 "James Forest" Lecture before the Institution of Civil Engineers. The tests were carried out by Mr. M. Atkinson Adam, B.Sc., Assoc. M. Inst. C.E. The first plant was adapted for a working load of 40 B.H.P. and the second for 30 B.H.P. In each case the producer was started up cold, and run on test for fully eight hours. At the start air was blown in by a small hand-power fan and after ten minutes from lighting up the gas was of a proper quality. The gas was then sucked through by a fan, which represented the action of a gas engine operating under a constant load sucking gas from a producer in the usual way. Thence the gas passed to a gas holder. Analysis samples were frequently taken and the anthracite analyses were undertaken by Mr. Bertram Blount, F.I.C., Assoc. Inst. C.E., whilst the gas analyses were carried out by Mr. Horatio Ballantyne, F.I.C. The heat efficiency of the producers was found in two ways:—

(1) Counting in the fuel used in the starting up operation which includes that necessary for the heating up of the plant.
(2) Omitting the first two hours of the test, and so giving the plant what may be termed a "flying start."

The quantities of water used are very interesting. The figures showed that for vaporization, the 40 B.H.P. plant used about 30 lb. per hour, whilst the 30 B.H.P. plant used about 20 lb. per hour. For the scrubber, the 40 B.H.P. plant used about 400 lb. per hour, and the 30 B.H.P. plant used about 380 lb. per hour. This shows how small a proportion of the total water consumption is needed for vaporization. The anthracite used was of an ordinary commercial kind, costing 14*s*. 6*d*. per ton at the pit, and about 24*s*. per ton delivered at Basingstoke. The efficiency figures for the two producer plants were found to be

"Standing start" .	40 B.H.P.	85 per cent.
	30 B.H.P.	75 per cent.
"Flying start" . .	40 B.H.P.	89 per cent.
	30 B.H.P.	86 per cent.

Reference should be made to the paper for detailed figures, but it may be mentioned that the gas was found on a general average to have a calorific value of 135 B.Th.U. per cubic foot, and have a composition as follows :—

H_2	15·5	per cent.
CH_4	1·2	,, ,,
CO	20·0	,, ,,
CO_2	7·0	,, ,,
O_2	0·5	,, ,,
N_2	55·8	,, ,,
	100·0	

120. Tests of Pressure Producers.—In 1904 some exhaustive tests were made in America on the results of employing different varieties of bituminous coal in pressure producer plants and in steam engines, and it is worth while to give a brief account * of some of the figures obtained (*see* p. 186).

In each case the output was about 200 E.H.P., and in most cases the length of the trials was from 10 to 30 hours.

Mr. Shober Burrows has reported the result of a 24 day test undertaken in 1906 on a **pressure producer plant** operating with bituminous fuel. Analysis of the fuel showed—

H_2O	14·68
Volatile combustible	30·98
Fixed carbon	42·93
Ash	10·08
S	1·33
	100·00
B.Th.U. per lb.	= 12,343.

* *The Times* Engineering Supplement, January 23, 1907.

Kind of Coal	Name of Sample	Coal burned per E.H.P. Steam Plant	Gas Plant	Ratio of Coal used by Steam Plant to that used in Gas Plant
		Lb.	Lb.	
Bitumin . . .	Alabama, No. 2 . .	4·08	1·64	2·49
Black lignite .	Colorado, No. 1 . .	4·84	1·71	2·83
Bitumin . . .	Illinois, No. 3 . .	4·34	1·79	2·42
" . . .	" " 4 . .	4·80	1·76	2·73
" . . .	Indiana, No. 1 . .	4·13	1·93	2·14
" . . .	" " 2 . .	4·35	1·55	2·81
" . . .	Ind. Terr., No. 1 .	4·04	1·83	2·21
" . . .	" " 4 .	4·64	1·43	3·24
" . . .	Iowa, No. 2 . . .	4·95	1·73	2·86
" . . .	Kansas, No. 5 . .	3·93	1·62	2·43
" . . .	Kentucky, No. 3 .	4·22	1·91	2·21
" . . .	Missouri, No. 2 . .	4·93	1·71	2·88
" . . .	W. Virginia, No. 1 .	3·90	1·57	2·48
" . . .	" " 4 .	3·62	1·29	2·80
" . . .	" " 7 .	3·55	1·46	2·43
" . . .	" " 8 .	3·63	1·78	2·04
" . . .	" " 9 .	3·46	1·40	2·47
" . . .	" " 12 .	3·53	1·50	2·35
" . . .	Wyoming, No. 2 .	5·90	2·07	2·85
		Average		2·57

The gas left the generator at about 644° F. and passed a water seal to the scrubber. Thence to a centrifugal tar extractor. The calorific value of the gas was found to be 156 B.Th.U. per cu. ft. and its composition was

CO_2	9·2
Ethylene	0·4
CO	20·9
H_2	15·6
Methane	1·9
N_2	52·0
	100·0

About 143 lb. of tar was extracted per ton of coal used in the producer, whilst the approximate figures show that an average of 1·39 lb. of coal was used per B.H.P.-hour. As this plant ran for 24 consecutive days without shutting down, it is evident that continuity of operation could be practically achieved.

The whole of these tests go to show the great fuel economy obtained by the use of gas plant as contrasted with steam-plant. Another feature in which the gas plant has the advantage is in the smallness of the **stand-by losses.** When a boiler is banked up for the night it consumes a very much larger quantity of coal during the period of banking than a producer plant of the same output would require. Actual measurements of this nature are recorded by Mr. Dowson in his book on Producer Gas, and it was found that in the case of steam power, the consumption of fuel per standing hour was 71·5 lb., and in the case of gas power, 3·5 lb. only, which shows a ratio of about 20 to 1. And since each of these figures is the mean of several tests, they are not open to the criticism that they represent isolated cases only.

It will be of advantage to record at this point what are the chief objects to be achieved in the design and working of producer plant—

(*a*) A fairly deep fuel bed should be allowed for, otherwise the air may blow through in thin places, and so lead to local variations in the temperature.

(*b*) Provision of some sort must be made to prevent caking or cavitation of the fuel.

(*c*) Fuel must be fed in and ashes removed in such a way as not to render the process discontinuous or intermittent.

(*d*) Leakage of gas from pressure producers must at all costs be avoided, as the gas contains a large proportion of poisonous CO.

There are a good many makes of pressure producer plant, and some are adaptable for by-product recovery. Among the latter one of the most prominent types is the Mond pro-

ducer, which is being used on a large scale in South Staffordshire. Here ammonia in the form of ammonium sulphate (Am_2SO_4) can be produced as a by-product and sold for a considerable amount—often more than enough to pay the coal bill. In this process, as has already been explained, the temperature of the producer must be kept low, and to do this, large quantities of steam are used, as much as 2½ lb. per lb. of coal. This has the effect of course of reducing somewhat the actual efficiency of the gas producer and of raising the percentage of hydrogen present, but not to such a point as to introduce trouble in a suitable engine.

121. Percentage of Hydrogen.—The percentage of hydrogen present in the gas to be employed in a gas engine regulates the amount of compression which can be used. A good compression is essential for high efficiency, but if the proportion of hydrogen is high the danger of pre-ignition has to be guarded against. The following table taken from a paper by Mr. J. R. Bibbins * shows the proportion of hydrogen present in various kinds of gas and the calorific value of the gas when taken alone, and when taken with its theoretically requisite proportion of air—

Gas	B.Th.U. per cu. ft.		H_2—per cent. by volume
	Gas	Mixture †	
Natural Pittsburg	978	91·0	3·0
Oil	846	93·0	32·0
Coal-gas	646	91·7	46·0
Carburetted water gas	575	92·0	40·0
Water gas	295	88·0	48·0
Producer, hard coal	144	68 0	20·0
,, soft	144	65·5	10·0
,, coke	125	63·0	10·0

Attempts have been made to reduce the proportion of

* "Fuel Gas for Internal Combustion Engines," *Cassier's Magazine*, 1906.

† Based on theoretical air for combustion. *See* also par. 143.

hydrogen by admitting some of the **exhaust gases** into the producer instead of water vapour. In this case the dissociation of CO_2 replaces that of H_2O. This process is called the "straight carbon-monoxide gas producer." It is claimed to work very well and to permit of very high compressions being used. The gas has a calorific value of 105 B.Th.U. and a composition of :—

CO	26·95	per cent.
H_2	0·20	,,
CO_2	1·75	,,
CH_4	0·50	,,
N_2	69·30	,,
O_2	1·30	,,

122. Comparison of Costs.—The following interesting comparison has been drawn up by Mr. L. Andrews * and is well worth study.

CAPITAL COST OF 16,000 K.W. PLANT.

	Steam Turbines	Gas Engines
	£	£
Engines and electric generators . .	96,000	161,700
Boilers, feed-pumps, coal handling plant, etc.	81,000	—
Producers, gas-cleaning and coal handling plant, with all pipes	—	77,700
Engine-room, building, cranes, and engine foundations	18,000	42,000
Switch-gear and wiring for ditto . .	5,250	5,250
	£200,250	£286,650
Allowance for contingencies, 5 per cent.	10,012	14,332
	£210,262	£300,982
Capital cost per K.W. installed . . .	£13·1	£18·88

* *Electrical Engineering*, October 24, 1907, and *S.A.*, January 30, 1908.

Running Cost on 100 per cent. Load Factor. Annual Output = 140,000,000 K.W. Hours.

	Steam Turbines	Gas Engines
	£	£
Fuel, 165,000 tons at 10*s*.	82,500	—
Fuel, acid, stores and repairs for producers, less sale of by-products . .	—	28,250
Labour	7,000	9,000
Repairs of turbine plant, including boilers, etc..	8,750	—
Repairs of gas plant (excluding producers)	—	6,000
Oil, waste, and stores (excluding producer stores)	1,750	4,370
Interest and depreciation at 10 per cent.	21,026	30,098
	£121,026	£78,118
Total cost per K.W. hour	0·204*d*.	0·135*d*.

Mr. Andrews also takes the case when the load factor is only 15 per cent., and in that condition of running the costs per KW.-hour came out at 0·545*d*. for steam turbines, and 0·566*d*. for gas engines. These rates are nearly the same, but with rise of load factor the balance would soon turn in favour of the gas plant. Mr. Andrew's estimate of the capital cost of the gas plant, viz. nearly £19 per KW., would now be considered unduly high.

123. The Use of Gas Plant for Marine Propulsion has been discussed before several engineering societies.

Mr. J. T. Milton in his 1906 paper before the Institution of Civil Engineers stated that he was led to give attention to engines of this kind in connexion with proposals to fit them in vessels classed with Lloyd's Register. The paper deals with engine problems only, and assumes that a proper and suitable type of producer capable of using cheap fuel would before long be available. The writer of the paper specifies the following conditions which must be satisfied by a successful marine engine—

(*a*) The engine must be reversible.

(*b*) It must be capable of being stopped quickly and of being started quickly either ahead or astern.

(*c*) It must be capable of being promptly speeded to any desired number of revolutions between dead slow and full speed, and of being kept steadily at the required speed for any length of time. "Dead slow" ought to be not faster than one-quarter of full speed, and should be less than this in very fast vessels.

(*d*) It must be capable of working well, not only in smooth water, but also in heavy weather, in a seaway in which the varying immersion of the propeller causes rapidly changing conditions of resistance.

(*e*) All working parts must be readily accessible for overhauling, and all wearing surfaces must be capable of being promptly and readily adjusted.

(*f*) The engine must be economical in fuel, and especially so at its ordinary working speed.

Certainly no existing engine complies with all these conditions, and reference should be made to Mr. Milton's paper for a discussion of the difficulties: some curves are there given showing the different turning moment curves for different arrangements of engine.

Another paper is that read by Mr. J. McKechnie before the Institution of Naval Architects * under the title "Propelling and Ordnance Machinery of Warships," and a portion of it deals with gas engine propulsion. It was stated that at the Vickers Works at Barrow-in-Furness there had been constructed internal combustion engines of a power equivalent to about 40,000 I.H.P., and that for three or four years almost continuous research work had been undertaken. As a result of the experiments a 2-stroke engine has been designed. This engine, it was claimed, could be worked by producer gas, oil, or compressed air, was reversible, and could take gas direct from a pressure producer without any scrubbing being necessary. To prevent the poisoning of the crew by the leakage of the gas from defective joints the pipes were jacketed with air under compression.

* *Proc, I.N.A.*, 1907,

Not only would the introduction of gas engines for warship propulsion lead to a gain of space and dead weight (so allowing the offensive or defensive matériel to be added to), but the better disposition of its parts, and the absence of funnels would admit of a great improvement in respect of an actual increase in the number of guns which could fire on either broadside. In the proposed plan of battleship construction, the gas producers are shown divided into two sets well on either side of the ship, and the propelling machinery is shown well aft. The deck is clear for gun barbettes. Mr. Milton gives the following comparative table illustrating the superiority of the gas engine plant so far as area occupied, weight and fuel consumption are concerned—

COMPARISON OF WEIGHTS, ETC., OF STEAM, GAS AND OIL MACHINERY FOR 16 000 H.P. BATTLESHIP

	Steam Engine	Gas Engine	Oil Engine
I.H.P. available for propelling the ship . . .	16,000	16,000	16,000
Weight of machinery, including usual auxiliaries, but not deck machinery	1,585 tons *	1,105 tons †	750 tons ‡
I.H.P. per ton of machinery	10·1	14·48	21·33
Area occupied by machinery: Engines and boilers or producers	7,250 sq. ft.	5,850 sq. ft.	4,100 sq. ft.
Area per I.H.P. . . .	0·453 sq. ft.	0·366 sq. ft.	0·257 sq. ft.
Fuel consumption in lb. per I.H.P. hour—			
At full power. . . .	1·6 lb.	1·0 lb.	0·6 lb.
At about ¼ full power .	1·66 lb.	1·15 lb.	0·75 lb.

A further paper is Mr. A. Vennell Coster's, before the Manchester Association of Engineers, dated 1907. As Mr. Coster had much experience with marine steam engines, and spent many years with Messrs. Crossley Bros., his

* Includes water in boilers.
† ,, ,, jackets and piping, but not coal in producers.
‡ ,, ,, jackets and piping.

experience gives his conclusions much authority. The following are the advantages claimed for the gas engine—

1. The ship driven with half the amount of fuel.
2. Standard losses reduced over 75 per cent.
3. Working pressure confined to the engine cylinders.
4. No boiler tubes or main steam pipes to burst, nor furnace crowns to collapse.
5. No priming in a heavy seaway, or water hammer in pipes and cylinders.
6. No more difficulties with the firing of boilers on a beam sea. Gas producers may be charged only twice every twenty-four hours, and the rolling and pitching of the vessel is rather an advantage than otherwise in assisting the fuel down from the charging hoppers.

The three main difficulties in the way are—

(1) The construction of a gas producer able to gasify all grades of bituminous coal.
(2) A simple method to cleanse the gas from tar, either before the introduction of the fuel into the producer proper; when in the producer; or after the gas has left the producer on its way to the engine.
(3) Perfect control of the gas-propelled vessel in starting, stopping, reversing and running at all speeds.

The first of these difficulties obviously is avoided if coke or anthracite is used in the producer, but this solution is neither economical nor satisfactory on other grounds. Bituminous coal must be regarded as the source of the power to be used for ship propulsion. Mr. Coster stated that in his scheme for the cargo vessel *Lord Antrim* the producers were worked by means of a down-draught at the top, and an up-draught at the bottom, which met at the centre and the gas was drawn off by suction. The gas was then thoroughly sprayed and cleaned by being passed through coke, sawdust and wool wood scrubbers.

The reversing difficulty can be met in small engines by the

use of a reversible propeller, but for obvious reasons this would not do in the case of large engines. For powers up to 500 H.P. gearing may be introduced to effect a reversal in the direction of propeller rotation, just as in a motor car, but this cannot be used when the power transmitted is really large.

One of the great difficulties in connexion with the utilization of the gas engine on board ship lies in the fact that when the speed of the ship is decreased, the resistance to motion is decreased at a far greater rate, and this means that the mean effective pressure on the piston must be capable of very considerable reduction. When an attempt is made to get very low mean effective pressures in a gas engine, the engine is liable to stop altogether—in fact the gas engine as at present devised is not sufficiently elastic in its manner of working to make it an effective rival to the steam turbine for marine purposes. The difficulty may be solved by driving generators from the gas engines, so producing electric current which can be used in motors driving the screw propellers, but this requires a great weight of machinery, and is costly.

124. The well-known firm of **Thornycroft** have been working a good deal at the problem of adapting gas engines to ship propulsion, and an illustration is shown in Fig. 59 of an engine they are interested in. The chief difficulty is to devise a suction producer which will work with bituminous or caking coal without the necessity of being provided with apparatus for the extraction of tar and other by-products. The tar often amounts to 4 or 5 per cent. and may be as high as 15 per cent. It is therefore necessary to arrange the producer so that all the tar produced is consumed before it leaves the producer. This can be done by feeding in the fresh fuel from below, so that the heavy hydrocarbons given off from it are consumed as they rise into the hotter part of the fire. To save weight and space Herr Capitaine has hit on the idea of cleaning the gas by introducing a fine water spray, which mixes with the dust and other impurities, making a kind of fog. This fog then passes into a centrifugal machine which is driven fast enough to throw out the impurities and leave clean gas in the middle, which is then drawn off by the engine. Mr. J. E.

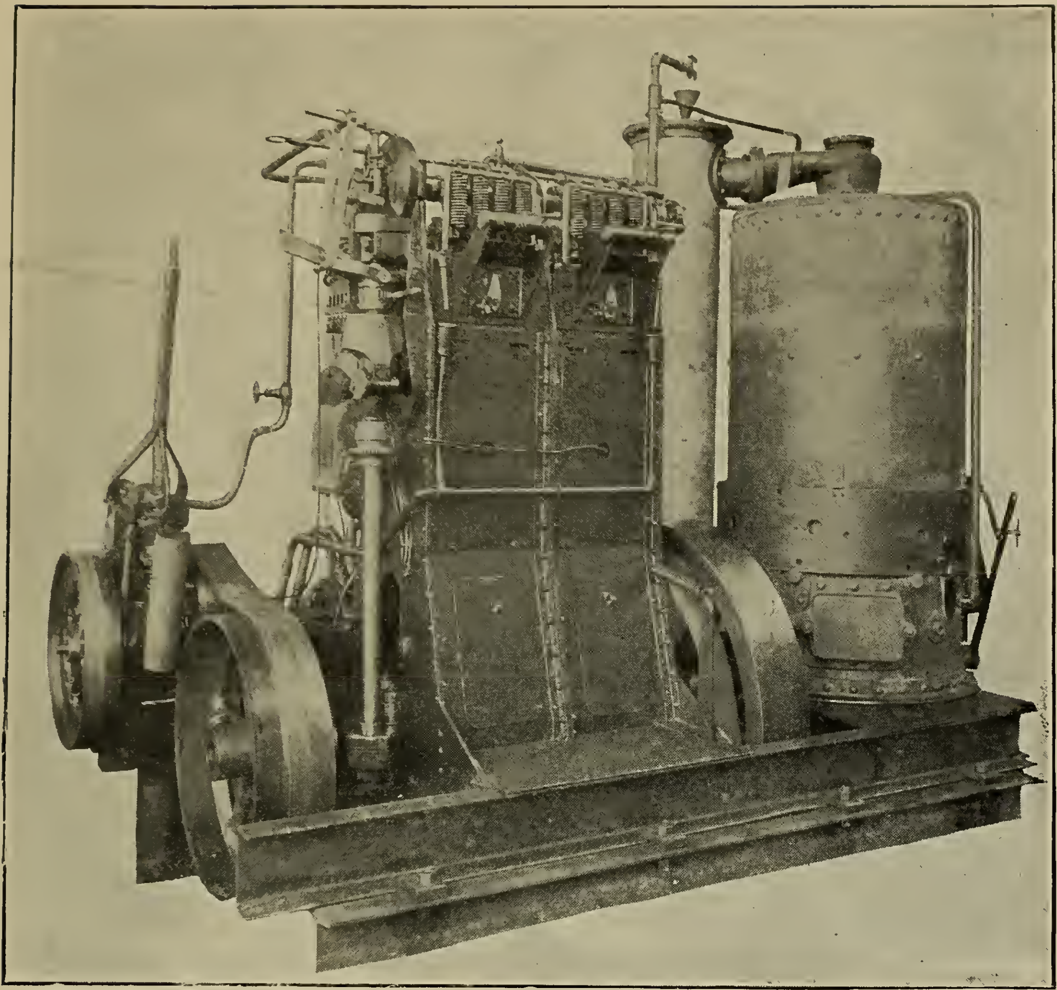


FIG. 59.—Two-cylinder Suction Gas Engine and Producer for Marine Purposes. (By courtesy of Messrs. J. I. Thornycroft & Co.)

Thornycroft * has given the composition of such gas as follows :—

CO_2	6	per cent.
CO	25	,, ,,
CH_2	1	,, ,,
H_2	14	,, ,,
N_2	54	,, ,,
	100	

He also remarks that "it will be realized that the size of the producer for a given power is comparatively small when

* Paper on "Gas Engines for Ship Propulsion," read April 5, 1906.

it is known that the area of the fire grate necessary is only 0·05 sq. ft. per H.P., whereas the average for an ordinary natural-draught steam boiler, burning 15 lb. coal per sq. ft. grate area, would be 0·2 sq. ft. per H.P."

The following test result is recorded by Mr. Thornycroft :—"Tests were made on November 8, 1904, with the *Gastug* No. 1 and *Elfriede*, a steam tug of very nearly the same dimensions and power. The *Gastug* No. 1 is 44 ft. 3 in. long by 10 ft. 6 in. beam, and is fitted with one of the four-cylinder 70 H.P. suction gas plants. The *Elfriede* is 47 ft. long by 12 ft. beam, and is fitted with a triple-expansion steam engine developing 75 H.P. At the towing meter the *Gastug* No. 1 attained a maximum pull of 2,140 lb., and the *Elfriede* a maximum of 2,020 lb. A run from Hamburg to Kiel and back was made by these two boats, during very stormy weather, at a maintained speed of 8½ knots. The consumption of fuel was measured for a period of 10 hours, and was as follows—For the *Gastug* No. 1, 530 lb. of German anthracite : for the *Elfriede*, 1,820 lb. of steam coal. This shows an economy of 1 to 3·44 in favour of the gas plant." Notwithstanding these successful efforts the difficulties to be solved before this method of ship propulsion becomes at all general are very great.*

125. Low Coal Engine.—An engine has been introduced by A. M. Low in which a special cylinder head contains a miniature gas producer, so that small coal can be fed direct to the engine. The coal passes down vertical tubes heated externally by the exhaust gases ; through these tubes is passed a stream of air and steam in the same proportions as in a suction gas producer. It is claimed that an experimental plant used only about 0·5 lb. of coal per B.H.P. hour, but detailed figures are not given of this test.†

APPENDIX A

The following is a description of the operation of a typical suction producer plant.

The suction type of gas-producing plant in question (**Campbell**) consists essentially of two main elements, a gas producer and a

* See *Internal Combustion Engineering*, p. 33, April 29, 1914.

† *The Engineer*, November 15, 1912.

gas scrubber. In addition to these there is a simple form of separator through which the gas passes on its way from the producer to the scrubber, and in which it deposits the heavier particles of dust which are carried over from the producer. A gas box is also provided between the scrubber and the engine, to act as a reservoir, from which the engine can draw a regular supply of gas.

1. *Method of Gas Production.*—In this method of gas production air and steam at atmospheric pressure are drawn through incandescent fuel by the motion of the engine, the oxygen, hydrogen and carbon combining in the producer to form a combustible gas which is suitable for power purposes. No boiler for providing steam under pressure is required, and no gasometer, the engine generating its supply of gas by the motion of the piston in the cylinder. The fuel used must be anthracite coal or coke (bituminous coal must not be used). The steam is generated in an evaporator which is heated by the fire in the gas producer. The air is drawn into the producer over the surface of the heated water in the evaporator and in passing takes up the steam which it then carries through the producer.

2. *General Instructions.*—The coal used should be passed through a sieve and no pieces under $\frac{1}{4}$ in. (5 mm.) should be used. The most suitable size is $\frac{5}{8}$ in. to 1 in. (16 mm. to 25 mm.). Coal dust is not only of no value, but it tends to stop up the pipes and interfere with the working of the plant. The fuel should not be imoistened before it is used. All the moisture required should be provided in the form of steam and pass through the fire in the ordinary way as described below. The evaporator should always be kept full of water to the overflow pipe. A water supply must be provided for the evaporator and the coke scrubber, and a drain to carry away the water from the scrubber. In any installation of this type, the engine should be erected as close to the producer as practicable so that the connecting pipes between the two are as short and direct as ean be arranged.

3. *To Start the Gas Producer after Erection or Cleaning.*—Before starting it is of the greatest importance to see that all the piping, cocks, and various vessels which go to make up the gas-producing plant *should be air tight*, as the apparatus when in operation is subjected to an excess of atmospheric pressure from without. If the various parts of the plant are not air tight, the air which leaks in will interfere with the quality of the gas and make it poorer. For this reason the whole apparatus should be tested after erection to prove the soundness of the joints, the test being carried out as follows: Referring to the illustration, if all the openings are elosed except the eock B, and air is then blown into the apparatus by the hand fan A, the various joints can be tested with a light. If air or gas escapes from the joints it will be at once detected. This test should be made periodically to see that everything is in order.

It may be carried out at any time after cleaning, and when everything is proved to be in good working order the engine should be made ready for immediate use when the gas is available.

Provided that all the joints are sound and tight, the water should now be turned on to the coke scrubber by means of the tap C until it overflows through the pipe D provided for that purpose. It is essential that the scrubber should contain sufficient water to seal the gas inlet.

Water should then be admitted to the evaporator F by means of the tap G until it just overflows in drops by the pipe T provided for that purpose. This overflow should be very slight before starting and must be regulated from time to time when running according to the load on the engine as described below. The taps C and G and the cock H should now be closed. The cock B and the cock J on the waste pipe K should be opened. The fire door L and the ashpit door M should then be opened and a fire of wood or coke started in the gas producer. *Ordinary bituminous coal must on no account be used.* When the fire is burning up well anthracite coal should be added through the hopper N in small quantities from time to time as the whole mass of fuel becomes incandescent throughout, this being continued until the gas producer is full to the level of the bottom of the gas pipe P. The fire door L and ashpit door M should be closed as soon as the coal is well alight. The hand fan A must be used for the purpose of blowing up the fire when starting, the whole of the products of combustion being blown by its means through the gas pipe P, separator R and uptake pipe K to waste, the cock J being open during this operation. Supposing the fire to have been lit for, say, fifteen to twenty minutes, and the hand fan to have been in operation during that time, the quality of the gas which is being made can now be tested by partially closing the cock J and thus passing the gas through the scrubber and gas box to the test cock Q. The nearer this test cock is placed to the engine the better; it can be placed at any convenient point in the gas pipe, between the engine and the gas box, for example. Before passing the gas through the scrubber the water must be turned on to the scrubber, by the tap C. The blowing will have to continue for a few minutes until the scrubber and gas box are cleared of air, and gas has been blown in to take its place. A light should then be placed to the test cock Q and the gas if of a good quality will burn with a steady flame. If the coal is of good quality the gas will burn with a long flame, orange red in colour, and one which does not go out. With some coals it is difficult to produce anything but a blue flame, but as long as the gas burns steadily it will generally be found that it is of sufficiently good quality to start the engine.

Caution.—When testing the gas, as described, care must be taken to turn the fan at a steady and even speed. Under no cir-

cumstances should the fan be stopped while the gas is burning at the test cock or the pressure will at once fall and the flame will

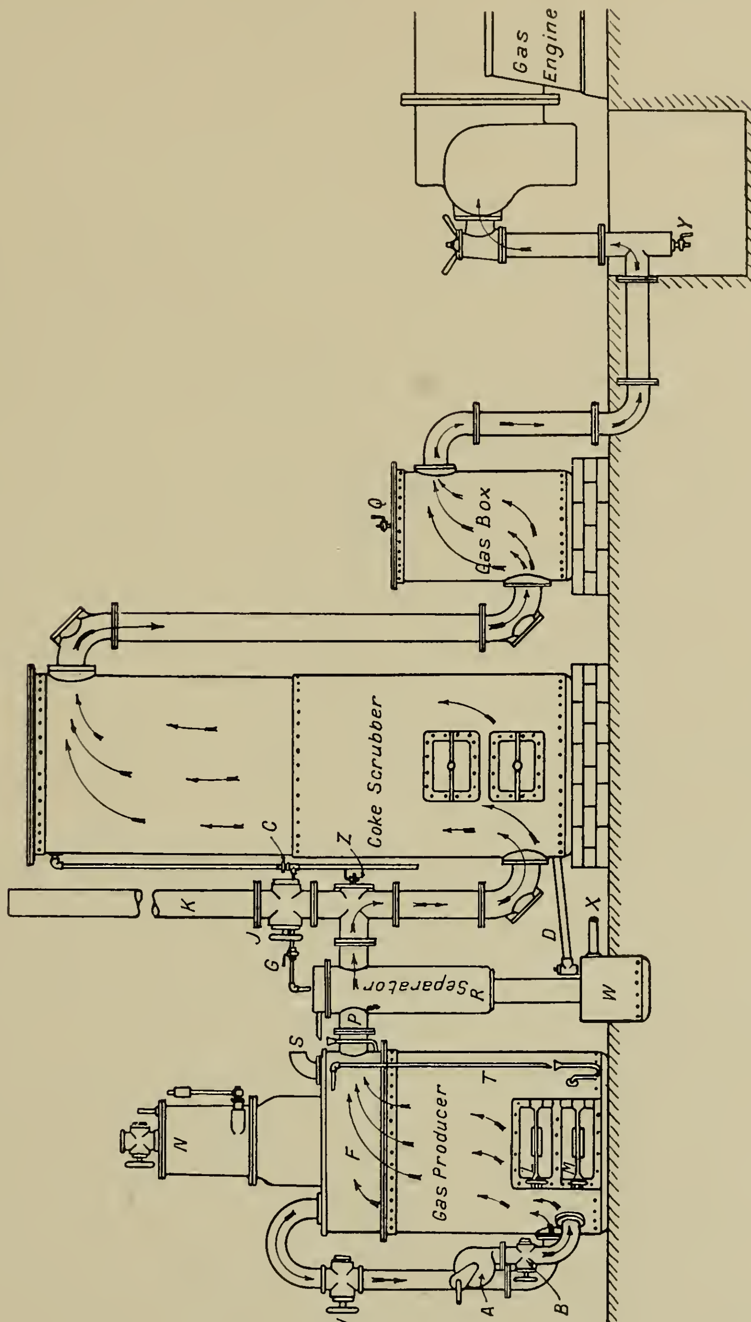


FIG. 60.—Diagram of Campbell Suction Gas Producer.

probably be drawn back into the gas box and fire the gas in the gas box and scrubber, the explosion caused thereby blowing the water out of the water seal and possibly doing other damage.

On the other hand the fan must not be blown too hard or the gas will be forced out through the water seal at the bottom of the coke scrubber.

The tap C should be opened to such an extent that the temperature of the lower portion of the scrubber does not rise above 100° Fahr. approx., the top of the scrubber being cold. The amount by which the tap G is opened must be regulated according to the load on the engine and so that the evaporator is always full and a slight surplus of water runs in drops only through the overflow pipe T into the ashpit when the engine is running under a full load. When running under a light load little or no water is required in the ashpit. An excess of water in the ashpit results in a poor quality of gas.

4. *To Start the Engine.*—As soon as the gas is burning satisfactorily at the test cock this cock and the cock J should be closed and the fan stopped. The engine should then be started in the usual way. No time must be lost in getting the engine started or the fire in the producer will become dull and a poor quality of gas be given off. Assuming that the engine has been started, the cock H should be opened and more water turned on to the scrubber by the tap C and to the evaporator by the tap G. The cock B should now be closed. By opening the cock H and closing B the air is drawn in through the inlet S and over the heated water in the evaporator F, the suction set up by the movement of the engine piston causing a constant indraught of air in the direction shown by the arrows in the sectional diagram. It may be mentioned here that the supply of air to the engine will have to be adjusted from time to time according to the quality of the gas. For this purpose a simple form of throttle valve should be provided in the air inlet passage through which air is supplied to the engine. This valve should be regulated so that as far as possible the engine takes in a supply of gas at every cycle and thus keeps the fire in the producer bright and in good condition. The engine should be provided with mechanism to ensure this being done.

5. *Method of Stoking the Gas Producer.*—The gas producer is provided with a hopper at the top for the purpose of feeding the fire. The hopper is provided with a swing door at the top and a valve with a weighted lever at the bottom so that when fresh coal is added the top door only is opened, the valve remaining closed. When the coal has been filled in through the hopper the top door is closed and the valve opened. By this means all air is excluded from the gas producer. Care should be taken to see that the valve to which the weighted lever is attached is properly closed so that no air can enter while the gas producer is working. Generally speaking it will be necessary to add a charge of anthracite every two or three hours ; this, however, must depend upon the size of the apparatus and the amount of power which the engine is develop-

ing. While the gas producer is in full operation the coal should not be allowed to fall below the lowest point in the evaporator. The top layer of coal should never be incandescent, this point can be watched through the mica window which is provided for that purpose at the top of the hopper. Previous to stopping the engine, however, the fire in the producer should be burnt down so as to leave only a moderate quantity of coal in the producer, sufficient to start up quickly again when required. How frequently the fire will have to be cleaned will depend upon the quality and amount of the fuel used ; speaking generally twice a day will be sufficient, once in the morning before starting and once at midday, if a stoppage is made then. Should it be necessary to stir up the fire whilst the engine is at work this can be done through a hole in the ash door by means of a poker. If it is necessary to take out the clinker whilst the engine is at work, this should be done very quickly so as to allow as little air as possible to enter the gas producer, for should an excess of air be allowed to enter, the gas would be of inferior quality. It is advisable as far as possible to leave the gas producer alone whilst the engine is at work, except for the occasional charges of coal which it requires. The gas producer should be cleaned out entirely about once a week and the clinker chipped off the firebrick lining of the producer if necessary. The producer should never be cleaned directly after the fire is raked out, but should be allowed to cool down gradually, otherwise the firebrick lining will probably crack through the rapid change in temperature.

6. *Hydraulic Box.*—The surplus from the coke scrubber is led into the hydraulic box W by means of the pipe D. This water forms at the same time a water seal for the pipe which connects with the separator mentioned above. The box W should be cleaned out every few weeks so as to keep it clear of the accumulated ash and small particles of coal which will come over with the gas. Special attention should be paid to the pipe D to see that no foreign matter settles in it. When the engine is at work the surface of the water in the hydraulic box will be in constant movement ; the movement, which should be a slight one, will vary with the amount of gas drawn away by the engine. An overflow pipe X is provided to run the water away from this box to a drain, or as may be arranged.

7. *Coke Scrubber.*—The coke scrubber is provided to remove from the gas all its impurities and at the same time to cool it. When the apparatus has been erected the inside of the scrubber should be thoroughly cleaned and the grating put in through the upper manhole. The scrubber should then be filled with well washed foundry coke, the size being not less than about 1 in. (25 mm.). The bottom layer of coke for a depth of about 8 in. (0·2 m.) should consist of pieces which are under any circumstances so large that they will not fall through the grating. The scrubber can then be

filled up with coke to about 4 in. (0·1 m.) below the water pipe. Before starting the bottom of the scrubber should be cleaned out through the doors provided for that purpose. All the openings in the scrubber should now be closed and the water supply turned on so that the coke is washed thoroughly free from all the particles of dust which it may contain. Every three or four weeks the bottom door of the coke scrubber should be opened to see whether there is any accumulation of dust in the form of mud at the bottom of the scrubber; this if present should be removed. When the coke is first put into place this examination should be made more frequently, as new coke frequently contains a large quantity of dust. The coke in the scrubber will, generally speaking, be serviceable for a period of nine to twelve months, but this depends upon the amount of work which the plant has to do. When it is found necessary to renew the coke in the scrubber the whole of the apparatus must be stopped, the waste pipe opened and all the ash and fire hole doors opened and left open for several hours before any work is done to the plant. The upper cover of the scrubber should then be removed and the coke taken out through the upper side door in the scrubber. This cleaning should take place during the daytime so that no fire or light need be brought into the gas-plant house while it is going on. The windows of the house should be open during the process of cleaning so that there is plenty of ventilation. It is advisable that there should always be two men present during the operation of cleaning, in case one of them should be overcome by the presence of gas. When replacing the doors on the scrubber after having renewed the coke care must be taken to see that the joints are sound and tight as already described.

8. *Piping and Gas Box.*—These should be looked to and cleaned about once a month. Impurities will settle in any pockets or where the course of the gas is not direct. For this reason all bent pipes should be avoided as far as possible and when present should be examined from time to time. The moisture which condenses in the gas box and in the pipe leading from it to the engine should be emptied out daily, otherwise it will get into the engine and interfere with its working. A drain cock should be provided, as at Y, for the purpose of drawing off this moisture.

9. *To Stop the Gas Producer.*—The gas cock on the engine should be shut and the waste cock J opened so as to allow the remaining gas to escape. The taps C and G and the cock H must then be closed and the ash door opened a few inches so as to allow the fire to continue burning.

10. *To Start the Apparatus again after a Temporary Stoppage.*—The fire and ash doors should be opened to clean the fire, any cinders or clinker should be removed without disturbing the fire as far as this is possible, the doors should then be closed, the cock

B opened, and the fan started until the fire is again in good condition. Anthracite must then be added until a good quality of gas is obtained, when the engine may be started up to work. When the stoppage is only temporary, the scrubber and gas box will probably be full of good gas when it takes place, and it is therefore better to test the fresh gas, made at restarting, by means of a test cock placed at Z rather than to test it at Q. When good gas is obtained at Z the cock J can be closed and the gas then sent through the scrubber and gas box to the engine. By following this plan the good gas remaining in the scrubber and gas box when the plant was stopped will be utilized, instead of being blown away to waste as might otherwise have been the case.

Caution.—The regulation of the supply of water to the coke scrubber is important. If the supply be too small, steam will be formed in the scrubber, the gas will not be properly cleaned, and the quality of the gas will deteriorate. If the supply be too great, the water seal of the gas pipe will be too deep and the engine will not be able to suck the gas through the producer. The coal should not be too large or of unequal size, or the air spaces between the various pieces will be too great. The guiding principle in this is to have a mass of fuel in the producer which is as homogeneous as possible without being solid. Where coke is used as the fuel a sawdust scrubber is required between the coke scrubber and the gas box. When a gas plant has been designed for anthracite, other modifications may be necessary if it is decided to change from anthracite coal to coke.

EXAMPLES

1. The following measurements were made during a test of a gas engine using producer gas : Volume of gas used per hour = 1,400 cu. ft. Calorific value of gas = 90 C.H.U. per cu. ft. B.H.P. = 29·2. Water flowing through jackets = 70 gallons per hour. Rise in temperature of jacket water = 60° C. Calculate :

(i) the number of C.H.U. supplied to the engine per hour.
(ii) the number of C.H.U. turned into useful work per hour.
(iii) the number of C.H.U. absorbed by the jacket water per hour.

How much heat per hour is unaccounted for, and what has become of it ?

2. A coal has the following analysis : carbon 88 per cent., hydrogen 4 per cent., oxygen 2·4 per cent., sulphur 1 per cent., the remainder being ash. Calculate the calorific value per lb. and the theoretical

quantity of air required for its complete combustion. State how you would ascertain the quantity of air actually supplied. (Cal. values, C=8,130; H=29,100; S=2,240.)

(B. of E., 1912.)

3. On a trial of a gas producer and engine the following particulars were noted :—

Duration of trial, 24 hours.
Total coal used, 2·4 tons.
Calorific value of coal, 14,500 B.Th.U.'s per lb.
I.H.P. of engine, $259\frac{1}{3}$.
Jacket-cooling water used per hour, 185 lb.
Temperature of jacket-cooling water :—inlet 60° F.
outlet 140° F.

If the thermal efficiency of the producer is 80 per cent., and assuming no loss between producer and engine, estimate the thermal efficiency of the engine and the percentages of total heat of combustion lost :—

(i) in the jacket-cooling water,
(ii) in the exhaust gases, by radiation, etc.

4. Suppose that for 1·2 lb. of coal we get 1 B.H.P.-hour from a gas engine using Dowson gas. This works a reversed heat engine, the mechanical efficiency of the engine being 85 per cent. Calculate the heat units added to the air per lb. of coal per hour and compare it with direct heating. What is the ratio of the amounts of air that can be heated over the same temperature range by the two processes. The calorific power of the coal is 8,200 C.H.U. per lb.

CHAPTER VII

Blast-Furnace and Coke-Oven Gases

Thermal Value—Cleaning the Gas—Utilization of the Surplus Power.

126. The Production of Waste Power from Blast-Furnace and Coke-Oven Gases.—The plan of using blast-furnace and coke-oven waste gases in gas engines is now quite largely followed; and the extent to which it may be put into force in any country depends chiefly upon that country's output of pig-iron. The following figures show the output in pig-iron in metric tons for the three countries chiefly concerned—

	1911
U.S.A.	23,600,000
Germany	15,300,000
Great Britain	9,700,000

The gas that issues from blast furnaces is rich in **carbon-monoxide** and poor in **hydrogen**, and has a calorific power of about 90 B.Th.U.* per cu. ft.: whereas the gas from coke ovens is extremely rich in hydrogen and may have a calorific value as high as 500 B.Th.U.† per cu. ft. The former is the easier to deal with as it is produced at a steadier rate, whilst with the small quantity of hydrogen which it contains, pre-ignitions are unlikely. Consequently it is safe to raise the

* 50 pound-calories.

† About 280 pound-calories.

compression to a much higher point (180 lb. per sq. inch or more) than would otherwise be safe, and the engine is thereby rendered of higher thermal efficiency. Both gases require cleaning in order to remove the dust.

127. Blast-Furnace Gases.—The idea of burning blast-furnace gases directly in gas engines instead of under steam boilers, as had previously been done, was first put into practice about the year 1894, nearly simultaneously in Great Britain, Germany and Belgium. The pioneers, prominent among whom was the late Mr. B. H. Thwaite, experimented with small engines and, as satisfactory results were obtained, it was soon desired to increase the scale of operation. In Germany great progress has now been made and recently a number of large plants have been put in in this country and in the U.S.A.

The calculation as to the power available in this way in Great Britain may be made in the following manner. The pig-iron output for 1911 (for example) was, in round figures,

$$10{,}000{,}000 \text{ tons},$$

and it is well established that the residual gases from blast furnaces in Great Britain as well as on the Continent and in America, are capable when used in internal combustion engines of yielding about **27 H.P. per ton of pig-iron per day** (the figures given by various engineers are as follows: Greiner, 20; Bryan Donkin, 28; Max Rotter, 25; Thompson, 20; Rossi, 30 to 35). It follows that the whole output would be about

$$\frac{10{,}000{,}000}{365} \times 27 = \mathbf{740{,}000\ H.P.},$$

of which at present the greater part is going to waste.

The corresponding H.P. for the 15,300,000 tons of output in Germany would be 1,100,000 H.P., which agrees generally with Dr. Hoffmann's estimate of 1,000,000 H.P.

It is confidently calculated that in those countries where this development is in progress a saving of several shillings per ton will be made in the cost of producing iron. Several German firms, notably, have already found very favourable financial results to accrue.

Professor H. Hubert remarks * that in Belgium the honour of being first in the field belongs to Messrs. Bailly and Kraft, of the Cockerill Co. The patent taken out by the Company for this new application was dated May 15, 1895, and the first trials were made at the end of that year. They were made with a Simplex engine of 8 H.P., in which the clearance space had been reduced in order to increase the compression and to facilitate the ignition of the mixture. The gas cleaning was imperfect, and was carried out simply by passing it through two scrubbers four metres high. The engine is stated to have displayed perfect elasticity, and adapted itself to the variations of composition, pressure and temperature of the gases.

The following interesting table is taken from Professor Hubert's paper—

Engine	Date of Trials	Power		Calories used per I.H.P. Hour	Thermal Efficiency
		I.H.P.	B.H.P.		per cent
8 H.P. engine	1896	5·26	4	4,030	15·77
200 H.P. engine (single cylinder, single acting, constant admission)	1898	213·9	181·82	2,775	22·9
600 H.P. engine (as above) .	1900	825·8	670·0	2,520	25·2
200 H.P. engine (as above, except for variable admission).	1901	246·9	215·3	2,766	23·0
1,400 H.P. engine (double-acting tandem, variable admission).	1906	1,755	1,582	2,129	29·8

128. Coke Oven Gases.—Coke-oven gases are much richer in hydrogen than blast-furnace gases, and they are therefore much more liable to pre-ignitions. To avoid this danger, the compression is not taken so high, although this precaution unfortunately has also the effect of tending to reduce efficiency. On the other hand their thermal value is far higher, often more than five times as high. To illustrate this, the following typical figures are given :—

* Iron and Steel Institute, 1906.

B.F. gas :—24½ per cent. of CO ; 62 per cent. of N_2 ; 1¼ per cent. of H_2 ; Calorific value 86 B.Th.U.* per cu. ft.
C. Oven gas :—50 per cent. of H_2 ; 40 per cent. of CH_4 ; Calorific value 560 B.Th.U.† per cu. ft.

To calculate the possible output obtainable from coke-oven gases in this country is not difficult. Taking the 1906 output of pig-iron as 10,000,000 tons, the consumption of hard coke may be put as about 11,000,000 tons. To produce this quantity of coke about 15,000,000 tons of coal would be required, which on coking would give off about one-fifth of its weight in the form of gas, corresponding to about 500,000,000 cubic feet of gas per day. Assuming that a quarter of this is available as a surplus for use in gas engines, and that it is of the thermal value of 500 B.Th.U.‡ per cu. ft., the corresponding thermal energy is easily calculated. If the gas engines used have a thermal efficiency of 30 per cent., the following H.P. would be available :—

$$\tfrac{1}{4} \times 500{,}000{,}000 \times \frac{500}{24 \times 60} \times \frac{778}{33{,}000} \times 0{\cdot}30 = 306{,}000 \text{ H.P.,}$$

or in round figures **300,000 H.P.** This is an estimate for the English output. Dr. Hoffmann has estimated the German output as from 550,000 to 600,000 H.P. Not a little enterprise has been shown in Germany in harnessing this source of power, and action is being taken in this country to the same end.

The proportion of one quarter, used in the above calculation § as to the fraction of the gas available for the production of this surplus power, depends upon chemical problems, but it has recently been found that by raising the temperature of the air entering the ovens to 1,000 or 1,100° C. by means of regenerators, only 45 to 55 per cent. of the total quantity of gas evolved from the fuel is required for the work of heat-

* About 50 pound-calories.

† About 330 pound-calories.

‡ About 380 pound-calories.

§ M. Léon Greiner gives the following approximate rules for the amount of surplus power available for use :—(a) with blast furnaces, the continuously available H.P. is equal to the number of tons of iron made per month ; (b) with by-product recovery ovens, the continuously available H.P. is equal to the number of tons of coke made per week.

ing the ovens, so that practically half the gas would in that case be available for the production of power in gas engines. This idea has been worked out by Mr. Koppers, and at the Anna Colliery of the Eschweiler Mining Co., near Aix-la-Chapelle, there are reported to be six batteries of Koppers regenerator ovens, with a power station designed for the production of 16,000 H.P. from the surplus gas.

It is on record * that at the Wath Main Colliery, Wath-upon-Dearne, Rotherham, an installation of 30 Hüessener patent by-product coke ovens, erected by the Coal Distillation Co. of Middlesbrough—representing the Actien Gesellschaft fuer Kohlendestillation—has been put in. The plant is to produce 800 tons of blast-furnace coke per week, and there is to be available sufficient surplus gas and surplus waste heat to produce 300 H.P. of electricity from the 30 ovens, in addition to meeting the requirements for power for coal grinding, elevating, and by-product plants. There are other instances of similar enterprise whereby English firms, on discarding the old "**beehive**" type of oven, have been able to obtain large quantities of surplus power. Of course there are other by-products besides power produced from coke ovens, such as sulphate of ammonia, coal-tar and benzole.

129. The Shelton Iron Works have some Koerting Engines working on **coke-oven gases**, and it has been found † that when some coals are used a calorific value of over 600 B.Th.U.‡ per cu. ft. is obtained, although 400 is more common. In no case, however, is the quality constant during the whole period of coking. It usually decreases from about 450 to 350 during the operation. The gas passes through scrubbers, where the ammonium sulphate and other by-products are collected and most of the tar removed. The gas then is divided into two almost equal parts, one half going to heat the coke ovens, and the rest to the production of power. As the gas contains much hydrogen, naphthalene, and other highly inflammable bodies, it is liable to pre-ignitions, and the compression is kept down to 100 lb. per sq. inch, instead of the 140 lb. per sq. inch, which

* *Times E.S.*, April 17, 1907.
† *Engineering*, February 15, 1907.
‡ About 330 pound-calories.

would otherwise be customary. The mean pressure works out at about 75 lb. per sq. inch. On testing a new variety of fuel the following results were obtained: Thermal value of gas 381 B.Th.U.* per cu. ft., engines developed 1 H.P. per hour per 22 cu. ft. at full load, or a thermal efficiency of $\frac{1,980,000}{22 \times 381 \times 778}$ $= 0{\cdot}30$. The analysis of the gas was—

CO_2	3·55 per cent.
Olefines, etc.	5·18 ,, ,,
O_2	1·59 ,, ,,
Methane	27·82 ,, ,,
H_2	54·33 ,, ,,
N_2	3·16 ,, ,,

According to some figures in *The Engineer*, of 22 installations in Germany with a total output of 13,000 H.P. from engines working on coke-oven gas, no less than eleven, or half of them, do not find it necessary to clean the gas. One of them was stated to be using gas with 0·2 per cent. of sulphur without injurious effect on the iron.

130. Cleaning the Gas.—It has been found that the most effective way of cleaning the gas is by the action of a water fed fan. The gas passes through a centrifugal fan which causes the heavy particles of dust to fly outwards, and at the same time water is fed into the fan and broken up by the same centrifugal action. This water catches up the dust particles and passes with them to a sump. Perhaps the best known gas cleaner of this type is the Theisen Patent Centrifugal Central-flow Gas Washer, made by Messrs. Richardsons, Westgarth & Co. It is illustrated in Figs. 61 and 62. The Theisen machines are specially adapted for cleaning gas, and particularly blast-furnace gas, for use in gas engines and where a high degree of purity is required. When very hot and dirty gas has to be treated, it is considered advisable to instal a preliminary saturator before the washer, where the gas may be cooled and the heavier dust removed. In this way not only is the volume of gas to be cleaned reduced,

* 212 pound-calories.

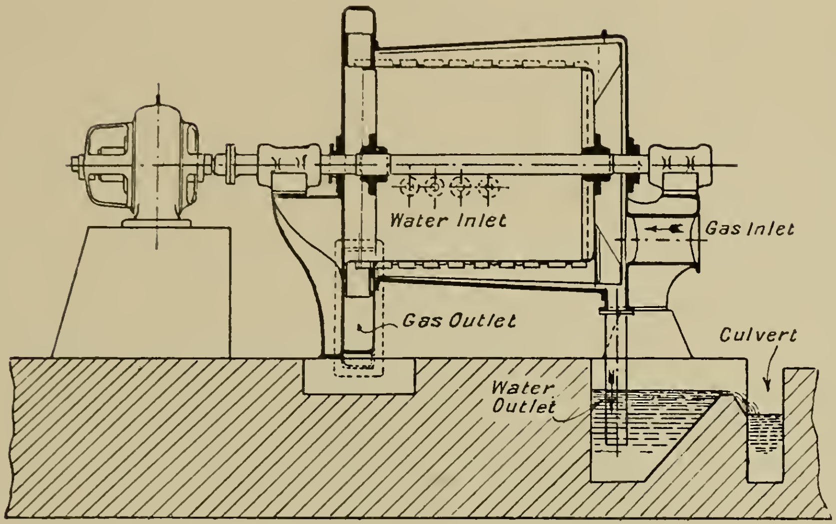


FIG. 61.—Theisen Gas Washer—Section.

but less water is required in the washer itself, and consequently less power is absorbed. The makers claim that the power taken to drive the cleaner does not exceed 2 per cent. of the maximum power which could be generated in gas engines from the gas cleaned. The quantity of water required by the Theisen apparatus varies with the temperature of the gas and the amount of dust therein, and in addition with the degree of cleaning necessary. With hot and dirty gas it sometimes happens that as much as 1 litre of water is required per cubic metre (or 1,000 litres) of gas cleaned, but usually half this quantity will suffice. Of course the water can be used again and again, if the dust be allowed to settle out of it. The makers have published the following table showing results of trials :—

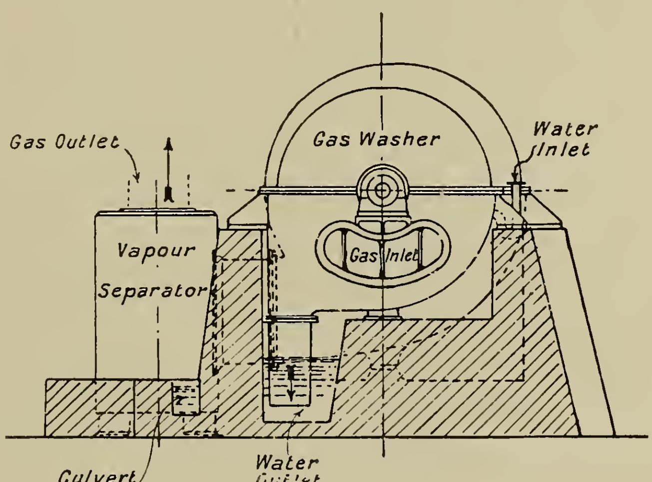


FIG. 62.—Theisen Gas Washer—End Elevation.

Results Obtained with Theisen's Apparatus Cleaning Blast Furnace Gas

Name of Ironworks	Quantity of Gas in Cub. met. per hour	Particulars of Gas						Water Used*			
		Entering			Leaving			Temperature		Quantity	
		Dust in grms. p. cu. met.	Temp.	Moisture, grammes p. cub. met.	Dust in grms. p. cu. met.	Temp.	Moisture, grammes p. cub. met.	Entering	Leaving	Gallons per hour	Gallons p. cub. met. gas
		Hot Gas direct from Furnaces									
Hochdahl . .	17,200	6	219° F.	17·8	·04	86° F.	7	57° F.	102° F.	4,163	·243
,, . .	12,000	6	316° F.	24	·02	99° F.	5	45° F.	104° F.	2,650	·22
Schalke . . .	10,200	3–4	291° F.	15 per cent. vol.	·004	86° F.	12–20	54° F.	131° F.	2,250	·22
		Cooled Gas with heavy dust separated									
Ormesby, near Middlesbro' .	7,500	2	126° F.	3	·0050	73° F.	18·5	58° F.	80° F.	448	·06
Hoerde . . .	12/15,000	2·5	115° F.	32	—	91° F.	3·45	82° F.	99° F.	2,650/3,530	{·23 ·234
,, . . .	6,000	2·34	113° F.	36·21	·01	82° F.	3·013	68° F.	93° F.	1,540	·253
Rombach . .	9,000	2	109° F.	42	·02	97° F.	3·2	64° F.	66° F.	2,250	·25

Mem.—1 Cubic Metre = 35·3 cubic feet.

The amount of dust in the gas can be measured very easily. It is only necessary to pass the gas through a filter consisting of a glass tube filled with absorbent cotton. The quantity of gas passed is measured in a meter, and the cotton is weighed before and after. The method is stated to give accurate results if the cotton is evenly packed along the tube and is not hygroscopic. In any case the cotton should be dried before and after in a desiccator, and weighed from time to time to check whether any moisture is held in it.

In America,* peculiar difficulties are experienced, owing to the character of the ores used. The Mesati ores are stated to be especially troublesome, owing to their friable nature. With every disturbance in the furnace great quantities of dust are evolved, which often pass the entire cleaning plant unless unusual precautions are taken. The suspended matter consists largely of ore dust together with some additional matter carried over from the other constituents of the furnace charge. At Bessemer, however, a cleaning plant has been put down which has cleaned the gas as low as 0·1 grain per cubic foot, which is considerably cleaner than the surrounding air in that particular locality. In practice, however, the engines work quite well with ten times this amount of dust.

131. Utilization of Surplus Power.—The utilization of the power derivable from the waste gases of blast furnaces and coke ovens is a problem in itself. The solution of this problem must depend upon the extent to which local demand for power exists, or can be created. It is only necessary to think of such electro-metallurgical processes as the manufacture of **aluminium** to bring to mind the possibility of the creation of huge demands for current under favourable conditions in respect of load factor. For the transmission of such power for any distance less than half a dozen miles, it would probably be most economical to use pipe lines to convey the gases, but for longer distances electrical transmission would be the obvious method to adopt. Another industry that might be served is the manufacture of **calcium carbide.** Carbide is not now being manufactured in bulk in this country, owing to the lack of cheap power. Abroad, engineers have

* *Times E.S.*, July 17, 1907.

the advantage of extraordinary cheap water power—as low, according to Professor S. P. Thompson, as $\frac{1}{26}$ part of a penny per H.P.-hour—and it is clear therefore that unless some very cheap source of power is rendered available here also it will not be possible for this country to produce its own carbide. Calcium carbide, in its purest form, is used for the production of acetylene for lighting purposes, but a less pure and cheaper kind can be used in the preparation of **chemical manure**, for which the demand is on an altogether larger scale.

Lime and coke when heated together to a temperature of 2,000–3,000° C. produce calcium carbide, combining in accordance with the following chemical formula—

$$CaO + 3C = CaC_2 + CO.$$

This reaction is carried out in an electric furnace worked either by direct or alternating current, although as the latter allows of a higher voltage transmission and simple transformation, it is usually preferred. It is a high temperature reaction and not an electrolytic one, thus permitting either type of current to be used. In the above equation the CO passes away as a by-product, and carries with it one-third of the carbon used. This gas might of course be collected and its thermal value used say for the heating up of the charge of lime and coke, for the earlier part of the great temperature range necessary. The amount of current needed to produce 1 ton of calcium carbide is about $\frac{2}{3}$ H.P.-year. Mr. Bertram Blount in his *Practical Electro-Chemistry* remarks : —" The surplus gas (from coke ovens and blast furnaces) can be used with economy in large gas engines of 500 or 1,000 H.P., and energy thus obtained almost as cheaply as from a water-power. For example, at an inclusive cost of $\frac{1}{10}d.$ per H.P.-hour, which is by no means unattainable, the price per H.P.-year is £3 13*s*., a figure which approaches that of a moderately cheap water-power. The real obstacle to the general utilization of such power is not its cost, but the somewhat restricted market for carbide, causing it to be readily swamped by any great increase of supply ; even with that restriction, however, the manufacturer having cheap coke

and lime in an industrial centre, will stand at least as good a chance as his rival with slightly cheaper power, but away from such supplies."

132. Owing to the discovery that calcium carbide could be used in the preparation of an excellent chemical manure, the possibility has been opened up of an enormous demand for this product, thus affording a suitable purpose to which large quantities of electric power could well be devoted. Such an enlargement of the calcium carbide market might not be altogether welcome to present manufacturers of the carbide, as the new product, not being used in the production of acetylene gas for lighting need not be so pure. A heavy demand for the less pure carbide might therefore lead to difficulty in obtaining small supplies of a purer kind, as it would hardly be worth while undertaking it. Or even if undertaken, the cost of such carbide might actually be greater with the increase of output than it is now. Probably if the bulk of the output were of a different quality it would not be feasible commercially to produce raw carbide in so pure a state, but this would not prevent the impurer carbide being purified by subsequent treatment in such quantities as the acetylene demand might necessitate. Even if chemical difficulties present themselves in the purification of the carbide when made, there is no reason to suppose that the ingenuity of chemists will be unable to circumvent those obstacles as soon as it is necessary for them to be dealt with. Present-day manufacturers hold to prevention being better than cure, and would far rather see that purer raw materials (coke and lime) were used; but if a big agricultural demand should arise, it is not to be expected that subsequent modes of manufacture would be controlled entirely with a view to the smaller market. The virtue of calcium carbide from the agricultural point of view lies in the fact that it can be converted into **calcium cyanamide**, which can be directly applied to land as a fertilizer, and that when so employed it is of great value and efficacy. The cyanamide can be obtained direct from the carbide by fusing the latter in a stream of nitrogen. Or if preferred, the process may be shortened by admitting nitrogen to the electric furnace in which the lime and coke are being fused.

EXAMPLES

1. A gas engine indicates 60 H.P. when using gas at the rate of 800 cu. ft. per hour. The calorific value of the gas is 300 C.H.U. per cu. ft. Calculate the thermal efficiency of the engine.

(B. of E., 1912.)

2. The following data are taken from a record of a test of a gas engine using power-gas :—cylinder diameter = 48 in., stroke = 54 in., M.E.P. = 75 lb. per sq. inch. Number of explosions per min. = 36. Gas used per min. = 1020 cu. ft. Calorific value of gas = 60 C.H.U. per cu. ft. B.H.P. = 545. Calculate :—

(i) The I.H.P.
(ii) The mechanical efficiency of engine.
(iii) Volume of gas used per I.H.P.-hour.
(iv) Volume of gas used per B.H.P.-hour.
(v) Indicated thermal efficiency.
(vi) Brake-thermal efficiency.

SECTION III

OIL AND PETROL ENGINES

CHAPTER VIII

Oil and Petrol Engines

Fuels — Slow-Speed Oil Engines — Diesel Engine — Petrol Engines for Motor Cars and Aircraft—Carburettors—Theory of Jet Carburettors—Ignition.

133. Fuels.—Internal combustion engines are of two classes : (1) those that work with gases for their explosive medium, and (2) those that use vapours of liquid hydrocarbons such as oils. The former class has been dealt with in the preceding chapters so far as everything except methods of ignition is concerned—and ignition being similar in both classes does not need to be dealt with in two parts. Oil and petrol engines, as those in class (2) are generally named, are of practically the same design as gas engines so far as cylinders, pistons, valves, etc., are concerned, and the difference between them mainly relates to the mechanism for dealing with the fuel used. A gas engine does not need any carburettor, whereas in the oil or petrol engine it is one of the most important and most sensitive parts.

The main requirements of liquid fuels for internal combustion engines are that they should

(1) Be moderate in cost ;
(2) Be free from anything which might lead to deposit inside cylinders ;
(3) Cause a minimum of difficulty when starting the engine ;
(4) Not lead to objectionable exhaust.

The liquid fuels in common use are heavy or medium oils, petrol, alcohol, and certain coal-tar products.

The crude petroleum coming from the well is treated at the oil refiners in such a way as to separate the light constituents from the heavy. This process is known as " distillation."

The crude oil is gradually heated in large boiler-like vessels, and as the vapours are given off they are led to a separate chamber and there condensed into fractions according to the boiling point or density.

The density of the oil on leaving the well is about 0·8 to 0·9, and the annual output for the whole world is about 50,000,000 tons. This total is very small when compared with the output of coal (which is more than 20 times as great), and unless extensive new supplies of oil are discovered it is useless to expect to replace solid by liquid fuel, although the higher calorific value of the latter might make the change desirable. The sources of the supply of petroleum in 1912 were as follows:—

Russia	18·0	per cent.
U.S.A.	62·9	,,
Dutch Indies	3·7	,,
Rumania	3·4	,,
Galicia	2·3	,,
India	2·1	,,
Other countries	7·6	,,
	100·0	,,

It will be seen that the first two countries produced 80 per cent. of the whole.

Pennsylvanian petroleum consists of a mixture of hydrocarbons of the C_nH_{2n+2} group in which n may be anything from 1 to, say, 30, and the boiling point rises gradually from 0° C. for C_4H_{10} to 280° C. for $C_{16}H_{34}$. Hexane (C_6H_{14}), which has a boiling point of 69° C. and a density of 0·667, is practically a light petrol; but the heavier petrols come nearer to octane (C_8H_{18}), which boils at 125° C. and has a density of 0·718.

134. Petrol.—Petrol* (called "gasoline" in U.S.A.) is one of the lightest of the constituents of petroleum put to commercial use. It begins to distil at 50° C.; at 80° C. to 90° C. about 50% comes over, and by the time the temperature has been raised to 150° C. all of it has been separated. This temperature range is 100 degrees Centigrade and is called the "distillation range." Petrol is an excellent fuel for motor-cars, having all the desir-

* One of its commercial names is "Benzine."

able characteristics. In 1902 the quantity imported into the United Kingdom was less than 6,000,000 galls.; in 1913 it was over 100,000,000 galls.

In the early days of motoring petrol was usually of about 0·68 specific gravity. Now, however, it is nearer 0·74. The best quality of petrol has a specific gravity of from 0·715 to 0·730 and yields 63 per cent. on distillation at 100° C. and 90 per cent. at 120° C. The calorific value is 10,800 pound-calories (19,500 B.Th.U.) per lb. (This is the "net" calorific value, as it does not include the latent heat of condensation of the water vapour formed by combustion.)

As is indicated by the large distillation range, petrol is not a single homogeneous product; but its *average* composition corresponds fairly closely to the chemical formula C_8H_{18}.

135. Paraffin.—Paraffin (called "Kerosene" in U.S.A.) is the next constituent to distil over. Its distillation range is much greater than that of petrol, being from 100° to 300° C. It is therefore even less of a homogeneous product than is petrol, and it is, moreover, not possible to draw a distinct line between the two, the one merging into the other. The specific gravity of paraffin is about 0·81. Its calorific value (net) is 12,500 pound-calories (22,500 B.Th.U.) per lb. Its average composition corresponds to $C_{10}H_{22}$. Owing to its large distillation range carburation is more difficult than with petrol, and it is seldom used therefore for engines of the motor-car type. After the paraffin has distilled over the remainder is separated into lubricating oil, residual oil, vaseline and paraffin wax.

136. Benzol.—The coal-tar product which is found to be suitable for use in internal combustion engines is benzol. It is a colourless liquid having a specific gravity as high as 0·88. The approximate chemical formula is C_6H_6* and its calorific value is about 11,000 pound-calories (19,800 B.Th.U.) per lb. It begins to distil at 80° C. and ends distillation at 120° C., showing that it is a more nearly homogeneous product than either petrol or paraffin. Owing to its high specific gravity it has a much higher calorific value per *gallon* than petrol. Its volatility renders it quite suitable for use in motor-cars, and

* "Benzene," the chief constituent of benzol; another—less important—constituent is toluene, C_7H_8.

it can be employed without any changes whatsoever in engine or carburettor. Some qualities have the disadvantage that at 0° C. they freeze solid, but the presence of a proportion of toluene in the benzol removes this difficulty.

137. Alcohol.—Alcohol is a volatile and colourless liquid of vegetable origin with a chemical formula of C_2H_6O.* Its specific gravity is 0·80. It distils completely between 80° C. and 100° C., and the calorific value is 7,000 pound-calories (12,600 B.Th.U.) per lb. It is, however, little used for fuel in England owing to the high Government duty.

Efforts have been made to encourage the use of alcohol in internal combustion engines, because of the cheap rate at which it can be manufactured on a large scale and because if produced here this country would be rendered independent of supplies of motor fuel from overseas.

138. Tabular Statement.—It is convenient to summarise the more important data in the form of a table

Liquid Fuels

Substance	Specific gravity at 15° C.	Distillation range deg. Cent.	Calorific value (net) C.H.U. per pound	Calorific value C.H.U. per Imperial gallon	Approximate Calorific value in ft.-lb. per pound
Paraffin . ($C_{10}\ H_{22}$)	0·81	100/300	12,500	102,000	18,000,000
Petrol . ($C_8\ H_{18}$)	0·73	50/150	10,800	79,000	15,000,000
Alcohol . ($C_2\ H_6\ O$)	0·80	80/100	7,000	56,000	10,000,000
Benzol . ($C_6\ H_6$)	0·88	80/120	11,000	97,000	15,000,000

139. Alcohol and Benzol Compared with Petrol.—The Fuels Committee of the Motor Union in their 1907 Report dealt largely with this matter. The following extracts are given—

"(1) *Safety.*—In the first place, in case of possible conflagration, alcohol can be extinguished by water, whereas petrol is only scattered under similar circumstances, and the area of

* Known chemically as Ethyl Alcohol.

conflagration increased. In the second place, and even more important, the flash point is considerably higher, being 60° Cent. compared with petrol, which may be taken as anything down to 10° Cent. below freezing point. This enables the alcohol to be carried and stored with safety under conditions where petrol would not be permitted. This further very much reduces the cost of freight and insurance.

"(2) *Thermal Efficiency.*—Owing to less air being required and a consequent reduction in the amount of inert gas, the thermal efficiency of alcohol is as high as 35 per cent., as against something below 20 per cent. in the case of petrol, and this greatly reduces the chances of overheating, besides also reducing the weight of cooling water, radiator, etc.

"(3) *Calorific Value.*—The calorific value of absolute alcohol is 12,600 B.Th.U., that of methyl alcohol with a specific gravity of 0·820 is 11,300, and alcohol with the addition of 20 per cent. of water shows a calorific value of 9,810; whereas that of petrol with a specific gravity of 0·722 ranges from 20,300 to 19,300 B.Th.U.

"(4) *Practical Limit of Compression.*—The practical limit of compression of alcohol is about **200 lb. per square inch**; and its explosion pressure is therefore considerably higher than that of petrol, the practical limit of compression of which—in view of possible pre-ignition—is limited to 80 lb. per square inch.

"(5) *Complete Combustion.*—With alcohol complete combustion is more easily attained, owing to the fact that it distils completely in its commercial form over a small range of temperature (80–100° Cent.), a very accurate degree of carburation thus being maintained. In the case of petrol the range of boiling point extends between 50° Cent. and 150° Cent.; such a large range of boiling points renders accurate carburation at all times more difficult, and makes the spirit what is commonly known as *stale* owing to the evaporation of the lighter fractions. Alcohol has not this disadvantage, the liquid being practically homogeneous throughout.

"(6) *Propagation of Flame.*—There is less rapid propagation of the flame when alcohol is used, which gives a much more uniform pressure throughout the stroke than petrol.

"(7) *Smell.*—With alcohol there is approximately no offensive smell in the exhaust, as compared with petrol.

"(8) *Flexibility.*—Alcohol will explode when mixed with air over a wider range than petrol—4–13 per cent. alcohol vapour in air being combustible, the range in the case of petrol vapour being 2–5 per cent.; thus the engine will be much more flexible.*

"There are three points, however, on which it is popularly supposed that alcohol compares unfavourably with petrol. These are:

(9) Corrosive effect
(10) Starting from cold
(11) Vaporization.

"(9) *Corrosive Effect.*—With regard to alcohol, any corrosive effect that may occur is probably due to impurities in the denaturing agent present in acetone and methyl alcohol, but these difficulties would be overcome if the carburation is such as to give complete combustion. Upon this point Dr. W. R. Ormandy writes to the Committee as follows:

"'My information with regard to the action of the effluent gases from motors running on alcohol was obtained from the engineer at the Gährungsversuchsanstallt at Berlin, who reported that engines running on pure alcohol, or even on pure alcohol with the German denaturant, gave no appreciable corrosion except on such parts of the motors as were so cold that condensation took place: thus the silencer was apt to corrode, more so the larger the percentage of water in the alcohol employed. As the average amount of water at present in German industrial alcohol is 10 per cent., this corrosion might become appreciable if the cooling of the cylinder walls was too effective. It has been proved, however, that the efficiency of alcohol engines is enormously increased by keeping the cylinder walls near the temperature of boiling water, and under these conditions no condensation and no corrosion obtained.'

"(10) *Starting from Cold.*—As for difficulty in starting from cold, it will be probable that alcohol as a fuel will almost always have a greater or less quantity of benzol mixed with it, in which

* This is more accurately explained in par. 142.

case this difficulty entirely disappears. Even without the addition of benzol there is little doubt that the question of starting from cold will be almost entirely overcome by the use of a suitable carburettor.

"(11) *Vaporization.*—Alcohol requires $5\frac{1}{2}$ per cent. of its total heat of combustion to vaporize it, whereas, on the other hand, petrol vaporizes without any external assistance. With regard to the heat required to vaporize it, it is to be noted that, inasmuch as a large amount of the heat produced passes off in the exhaust, this is really available for the purpose of vaporization and does not represent any thermal loss.

"*Other Means of Utilizing Alcohol.*—From the previous argument it will be seen that, in order to utilize alcohol in an internal combustion engine, certain modifications in the engine itself become necessary, but it is quite reasonable to expect that such alterations would be unnecessary if the proportion of tar benzol, acetylene, or other hydrocarbons containing a high percentage of carbon were mixed with the alcohol. Owing to this high percentage of carbon present, the chemical composition of the mixture will be brought more nearly to resemble that of the petroleum products. As to the most suitable relative proportions, experiment only will determine these, but such a fuel as is here suggested has the advantage of being a home production, as well as one that could be used without material alteration to the engine.

* * * * * *

"It has been stated in evidence that the average price at which alcohol can be produced in Germany amounts to 1*s.* a gallon, including the cost of denaturing and Government supervision. It is also a fact that in this country the actual cost of manufacturing alcohol amounts to $11\frac{1}{2}$*d.* a gallon (64 overproof, a strength common in industrial spirit)—see Report of Departmental Committee on Industrial Alcohol. This is produced from beet, potatoes, and molasses. Evidence has been given which tends to show that alcohol may also be produced from sawdust at a very low cost. The lowest figure it is possible to touch in this respect is 3*d.* per gallon when peat is used. Now, owing to the great strictness of the Excise authorities in England, the cost of denaturing and expenses

of supervision bring the total cost of the alcohol up to about 2*s*. per gallon at the present time, and it is therefore evident that should the Government see their way to take a wider view of the question of alcohol as a fuel for internal combustion engines this price of 2*s*. a gallon could be very materially reduced. If this were done, the price could easily be brought to such a figure that it would be a very serious competitor with petrol in this respect alone.

"The Government that will recognize this, and will allow untaxed alcohol suitably denatured to be used for light, heat, or power, will be conferring an immense boon and benefiting a very large proportion of the population."

As regards the use of **benzol** the Committee remark:

"What is commonly known as 90 per cent. benzol can be utilized with perfect success in the engine of a motor car either alone or mixed with petrol, or mixed with alcohol. Owing to the high percentage of carbon which is found in benzol, and to the low percentage of carbon in alcohol, it is evident that a mixture of these two liquids more nearly approaches the ordinary hydrocarbon liquid fuels to which we are accustomed in its chemical composition. Benzol will carburate air in the ordinary way when an ordinary petrol carburettor is used, but its specific gravity is very much higher than that of petrol, viz. 0·883, which may necessitate an adjustment of the float to prevent the benzol standing too low in the jet of the carburettor. Crude benzol inevitably contains a certain amount of foreign matter in combination with sulphur, which imparts to it an unpleasant smell in the liquid state. Owing to its comparatively low price, however, it might pay to have benzol still further treated after washing in order to remove these impurities, which could be done for the expense of about 1*d*. per gallon. At the present time benzol cannot be obtained in very large quantities, as the number of recovery plants in this country is not very large. As benzol is a home production, its use should be encouraged, and particularly at this present time when the difference between the prices of petrol and benzol is very small.

"Mixtures of benzol and alcohol have been tried in a desultory manner on the Continent, but in this country nothing has

been done upon an extensive scale. The possibilities for the successful use of such a mixture are very great, and both these fuels are capable of manufacture in this country in very large quantities. Although a mixture of benzol and alcohol is in its normal state quite nauseous, and would not require a further treatment such as the addition of wood naphtha, yet it is possible, at any rate, to partially separate these two liquids, the alcohol having an affinity for water."

140. Sources of Petrol Supply.—The advent of the motor car has been the main cause of the increase in the demand for petrol, which previously had been regarded as a waste product. Petrol is now one of the most valuable components of crude mineral oil. According to Boverton Redwood and V. B. Lewes * the following are the leading sources of the petroleum spirit (or petrol) imported into Great Britain :—

From	1912 Per cent.
United States	20
Sumatra, East Indies, Borneo and Netherlands. .	57
Russia and Rumania.	11
Other Countries	12
	100

Processes have been tried and are to some extent in use in this country, for obtaining petrol from paraffin or heavier oils by the chemical process known as "cracking." There is a tendency, however for these processes to be kept secret.

141. Latent Heat of Vaporization of Liquid Fuels.—When water is raised to the boiling point and the application of heat is continued, some of the water is converted into steam ; each pound weight of water formed into steam at 100° C. needs 537 pound-calories to effect the conversion. This 537 calories is called the latent heat of vaporization of water, or sometimes the latent heat of steam. In the same way, heat has to be given to petrol, paraffin, benzol, or alcohol to vaporize them. When these fuels are being used in internal combustion engines

* Imperial Motor Transport Council, 1913.

they need to be vaporized, and the heat necessary for vaporization is drawn from the air and metal in contact with them. If this heat be not provided in some way the fuel in the carburettor will get so cold that it will not vaporize satisfactorily. The usual way of maintaining the temperature at a satisfactory level is either to warm the entering air by making it pass round the hot exhaust pipes, or by jacketing the carburettor with cylinder-jacket water. If the former plan be followed it is found that the air needs to be heated by the following amounts for the fuels mentioned :—

Petrol	about	25° C. (77° F.)
Benzol	,,	30° C. (86° F.)
Paraffin	,,	60° C. (140° F.)
Alcohol	,,	100° C. (212° F.)

142. Proportion of Air to Fuel.—The chemical composition of petrol varies with its density, but taking it at the average of C_8H_{18}, the equation for complete combustion with oxygen is

$$2C_8H_{18}+25O_2=16CO_2+18\ H_2O$$

showing that 1 volume of petrol vapour will need for its complete combustion $12\frac{1}{2}$ volumes of oxygen. And since air contains 23 per cent. of oxygen by weight, or 21 per cent. by volume, each volume of petrol vapour will need about 60 volumes of air. It is also possible to calculate the proportion by weight instead of volume. Thus in the above instance 2 $(96+18)$ pounds of petrol need 25×32 pounds of oxygen and therefore $\frac{25\times 3200}{23}$ pounds of air. The number of pounds of air needed for the complete combustion of one pound of petrol is therefore $\frac{25\times 3200}{23\times 228}=15$ pounds.

Fuel	Proportion of Air to Fuel for complete combustion	
	Ratio by volume	Ratio by weight
Paraffin	74	15
Alcohol	14	9
Benzol	32	13
Petrol	60	15

The foregoing table gives also the corresponding figures for paraffin, alcohol and benzol. These are the proportions for complete combustion, but proportions varying within limits from these ideal figures will also explode. This is shown in the following table :—

Substance	Ratio by volume to air	
	Ideal	Practically possible
Alcohol	7 per cent.	4·0 per cent. to 13·6 per cent.
Benzol	3 ,, ,,	2·7 ,, ,, ,, 6·3 ,, ,,
Petrol	1·7 ,, ,,	1·0 ,, ,, ,, 5·0 ,, ,,

143. Calorific Value of Explosive Mixtures.—Although the calorific values of fuels vary so much—thus the calorific value of illuminating gas is many times as great as that of blast furnace gas—there is relatively little difference between the calorific volumes of the explosive *mixtures* formed with them. This is because the rich fuels are diluted with a great deal of air, and the poor ones with very little. The following table illustrates this :—

Fuel	Approx. calorific value per cu. ft.	Volumes of air theoretically needed for combustion	Approx. calorific value per cu. ft. of mixture
	pound-calories		pound-calories
Blast-furnace gas .	50	0·6	31
Producer gas . .	70	0·9	37
Coke-oven gas . .	300	5	50
Illuminating gas .	350	5	58
Alcohol vapour .	900	14	60
Benzol vapour . .	2390	32	72
Petrol vapour . .	3420	60	56
Paraffin vapour. .	4960	74	66

In practice there is always present some excess of air over and above the amount theoretically needed ; this leads to a reduction in the average calorific value of the gaseous mixture

and probably brings the figures in the last column of the above table still nearer to one another.

144. Slow-speed Oil Engines.—As an ordinary petrol engine which is run on paraffin becomes thereby an "oil engine," it is necessary to bring in the variable factor, speed, in order to distinguish it from the older and heavier types of oil engine. The former runs at, say, 1,000 revolutions per minute and over, and the latter at only a few hundred. Neither is essentially different from the other. If a petrol engine is imagined as greatly increased in size—say to a cylinder diameter of 14 in, —and all parts increased in proportion, the safe speed at which the engine will run will need to be reduced, because whilst the weight of moving parts goes up with the cube of the dimensions, sectional areas of stressed metal only increase as the square. It is not possible therefore to aim at high speeds without greatly increasing the cost of production. The building of such expensive engines is frankly put on one side and a cheap engine is built which will run at a very slow speed, slower even than the proportion to the increase in dimensions would naturally suggest. This enables cheaper materials to be used than are employed in the construction of petrol engines. The output in horse-power is reduced in proportion to the speed, but increased as the cube of the cylinder dimension provided that ports, etc., are designed of sufficient size to enable the working mixture to enter and leave the cylinder without undue obstruction.

As representative of this heavier class of engine the well-known **Campbell** oil engine is selected.

145. The **Campbell Oil Engine** is illustrated in Figs. 63 and 64. Fig. 63 shows the engine to be somewhat similar in plan to a horizontal gas engine and the engine parts are generally on that scale. The inlet valve C and exhaust valve G are shown in position. The latter is worked through a lever H and side rod J by an eccentric K driven from the crankshaft L by spur gearing. When the speed exceeds the normal, a centrifugal governor pushes down a steel piece N, which engages with a corresponding steel piece O on the exhaust lever H, and prevents the exhaust valve G from closing. When this valve is held open no partial vacuum can form in the

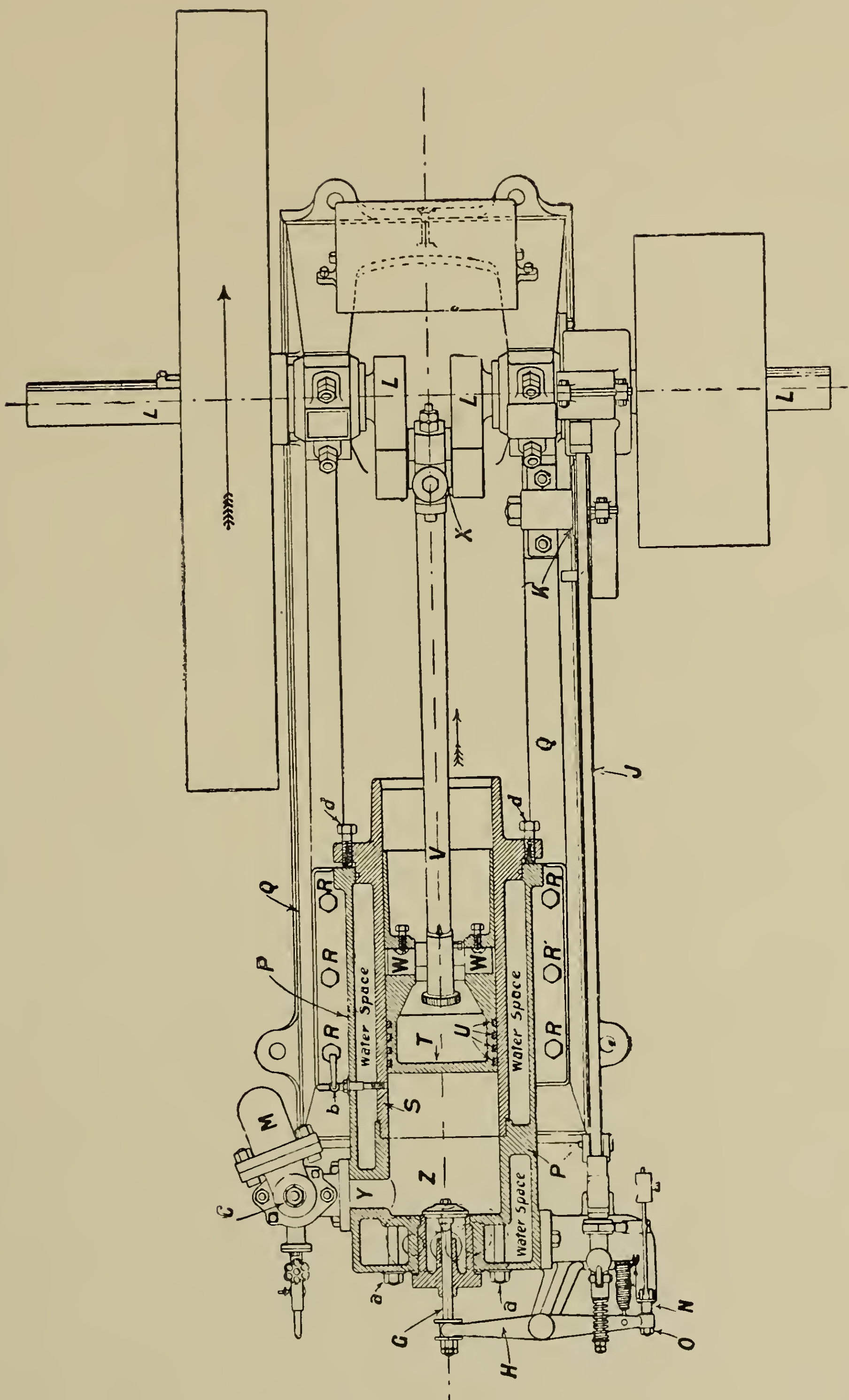


FIG. 63.—Sectional Plan through cylinder of Campbell Oil Engine.

cylinder during the charging stroke of the piston because there is free communication with the atmosphere through the exhaust valve, and consequently no charge of oil and air can be drawn into the cylinder. The **vaporizer** for combining air and oil into an explosive mixture is shown in section in Fig. 64 and consists of a cast-iron chamber A securely bolted to the

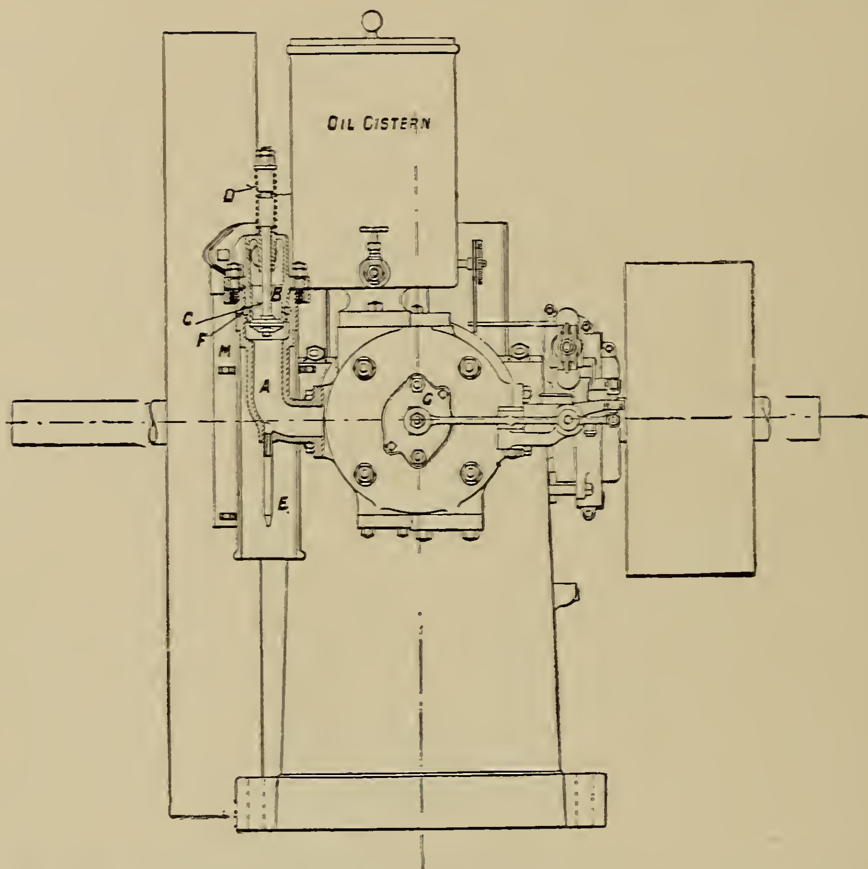


FIG. 64.—Campbell Oil Engine, illustrating operation of vaporizer.

cylinder and in direct communication with the combustion chamber. Into the top of this chamber the inlet valve plug B is fitted and this plug contains the seat of the inlet valve C (*see* Fig. 63). The inlet valve C is kept closed by a light spring D and only opens during the charging stroke of the piston when a partial vacuum is formed in the cylinder. Oil is admitted through the annular space or groove F, and passes through small holes in the valve seat and into the vaporizer when the inlet valve leaves its seat. Air is admitted through

the pipe M and passes through the inside of valve plug B, carrying the oil with it. The ignition tube E is screwed into a boss on the chamber A. The tube and the whole of the vaporizer is kept hot by an external lamp.* During the charging stroke of the piston, a partial vacuum is formed in the cylinder and the charge of oil and air is drawn through the inlet valve and sprayed against the heated sides of the chamber A. The mixture then passes into the cylinder, is compressed on the return stroke of the piston and then fired by the heat from the ignition tube.

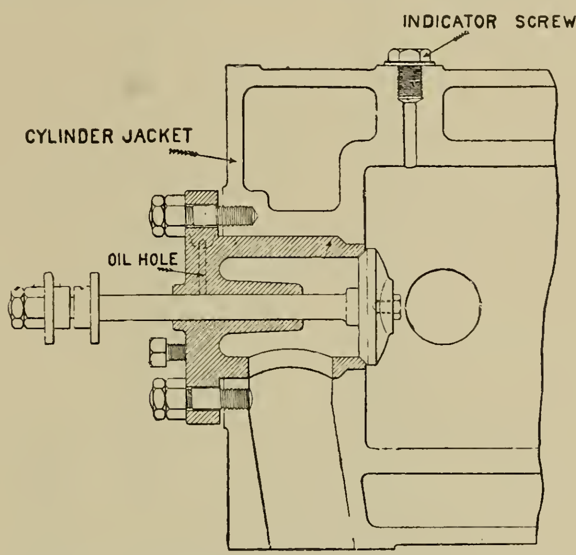


FIG. 65.—Enlarged view of Exhaust valve of Engine shown in Figs. 63 and 64.

146. The Hornsby Type of Vaporizer is also worth studying. This type of vaporizer is shown in Fig. 66 and to show how the vaporizer is fitted in place the diagram includes the cylinder also. When it is desired to start the engine a lamp is placed under the vaporizer chamber until the latter is at a sufficient temperature to ignite the oil which is pumped into it. This lamp is withdrawn once the engine is started, as the heat of explosion is sufficient to keep the temperature up to the requisite point. The oil tank is under the engine, and from it the oil is forced by a small pump into the vaporizer just at the moment when the piston is starting on its out-stroke and is drawing in the air necessary to combustion.† The supply

* In some later engines a large ignition tube is used which is able to retain enough heat from each explosion to fire the next, so that the lamp may be withdrawn once the engine has been started.

† In construction this type of engine is similar to the "Hot bulb" or "Semi-Diesel" engine. (See par. 147 for comparison with the "Diesel.") In these latter engines, however, the fuel is not injected into the vaporizer until the piston has reached the end of the compression stroke.

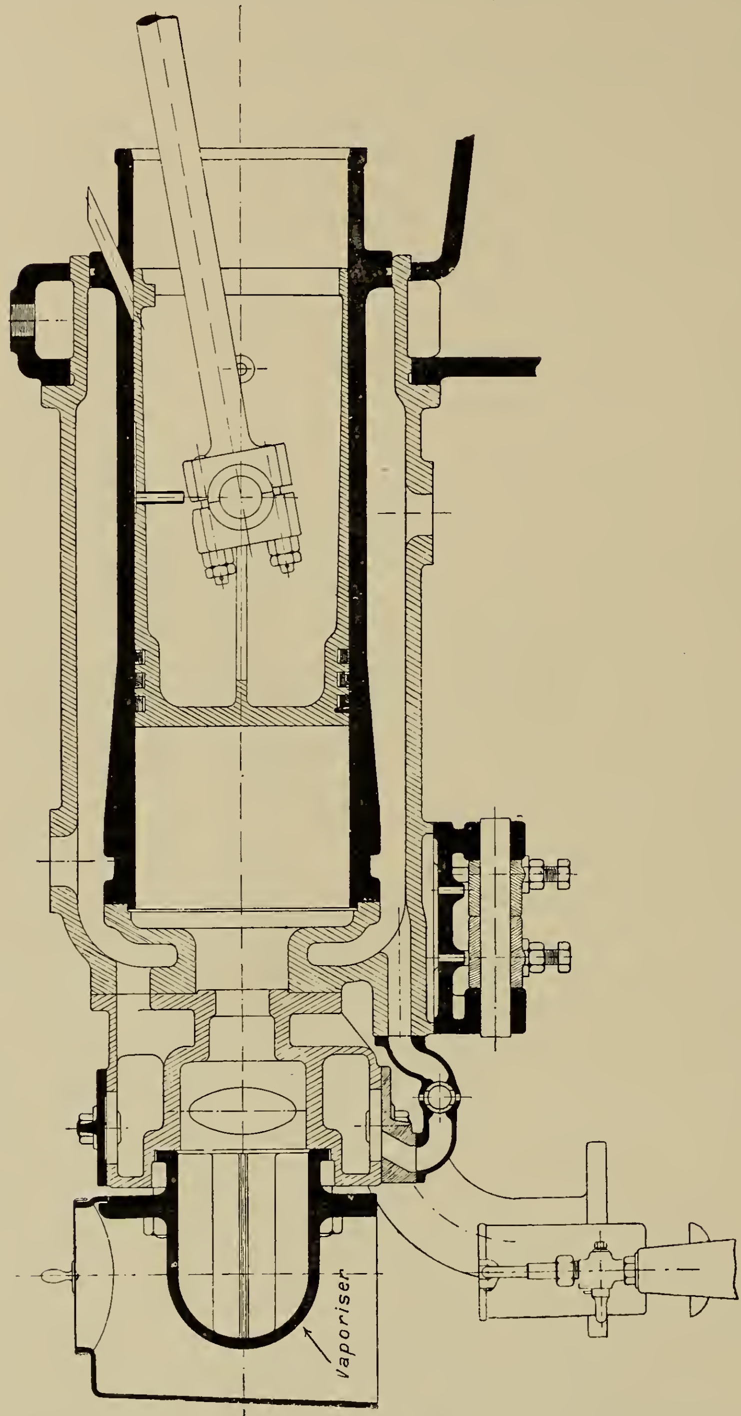


FIG. 66.—Section of cylinder and vaporizer—Hornsby Oil Engine.

of oil is controlled by the governor in the following way. The oil passes through a valve-box with two valves, one of which leads to the vaporizer and the other leads to an overflow

from which the oil can flow back to the tank. If the speed rises beyond the required point the governor opens this latter valve and the quantity of oil getting into the vaporizer is therefore reduced. On the return stroke of the piston the mixture is compressed and some of it forced back into the hot vaporizer, where the temperature is so high that ignition occurs and a working stroke is therefore made by the piston. The vaporizer chamber can, of course, be taken out and cleaned when desired. It is found, however, that even when working on quite heavy unpurified oils very occasional cleaning will suffice.

147. Diesel and semi-Diesel Engines.—Both these types of engine work with a heavy oil fuel. In both cases air only is admitted during the suction stroke, and in both the compression is carried to a much higher pressure and temperature than is possible in oil engines in which an explosive mixture of air and oil vapour are compressed together. The first Diesel engine was built at Augsburg * in 1897, and since then many thousand engines have been constructed for the smaller naval ships, for power generation and for other purposes. In this engine the compression is carried to 500 lb. per sq. inch (correspond-

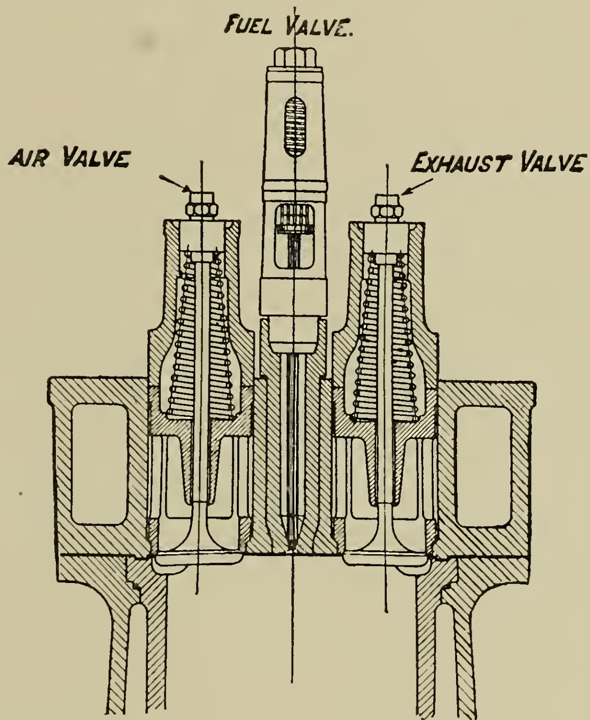


FIG. 67.—Cylinder Head and Valves of Diesel Engine.

* Dr. Rudolph Diesel on "The Diesel Oil Engine," *Proc. I. M. E.* 1912. Diesel's original idea had been to follow the constant temperature cycle, but the large size of the engine in proportion to the indicated power made the mechanical efficiency extremely poor and the cost very high; the plan was therefore abandoned.

ing to a compression ratio of about 12) and in the semi-Diesel to about 150 lb. per sq. inch ; whereas in the ordinary type of oil engine the compression cannot be taken much beyond 60 to 70 lb. per sq. inch (gauge pressure) without the charge pre-igniting. The compression space in the Diesel engine has to be exceedingly small ; and to facilitate this the arrangement at the head of the cylinder is as shown in Fig. 67. In the Diesel and semi-Diesel engines pre-ignitions of the ordinary kind are equally impossible, as there is no fuel present during any part of the compression stroke—although if from defective construction the oil inlet-valve of a Diesel engine should leak, there is a risk that explosion may take place

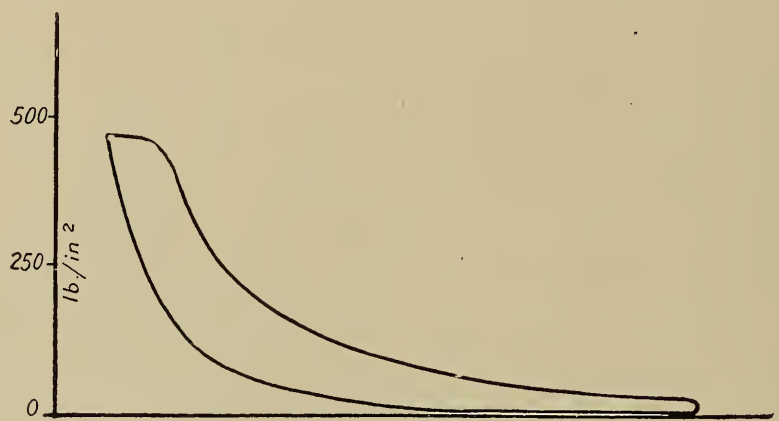


FIG. 68.—Indicator Diagram from Diesel Engine.

during the compression stroke and that the engine be damaged unless massively built. This danger is eliminated in the semi-Diesel engines, which have the oil supply forced into the cylinder by the action of a force pump which comes into action exactly on the dead point—with this arrangement the only result of a leaky valve would be leakage *from* and not into the cylinder.

In the Diesel engine the fuel is sprayed in through the fuel valve shown by air at 800 lb. per sq. inch as soon as the valve opens. The compression stroke has previously compressed the air and exhaust products to 500 lb. per sq. inch, corresponding to a temperature of about 600° C. The oil vapour therefore burns as it enters, and the pressure is maintained

at about 500 lb. per sq. inch during an appreciable portion of the outward stroke of the piston until at a given point the oil supply is cut off and expansion takes place. Fig. 68 shows an indicator card typical of this cycle. In computing I.H.P. from such a card, deduction needs to be made for work done in the air-compressor. Every Diesel engine is equipped with an air compressor for maintaining a supply of air in the starting and air injection vessels. The pressure is usually maintained in these vessels at from 800 to 1,000 lb. per sq.

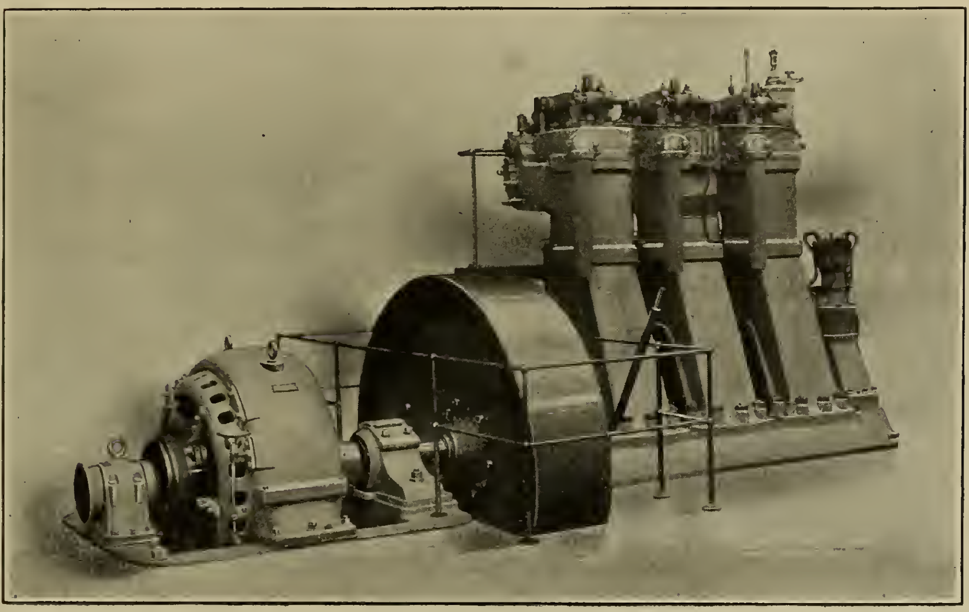


FIG. 69.—Three-cylinder "Mirrlees Diesel" Oil Engine coupled direct to 90 K.W. Generator. For Birkdale District Electric Supply Co.

inch and must not be allowed to fall below about 600 lb. per sq. inch, otherwise the compression pressure in the working cylinder would prevent the fuel entering and the engine could not be started.

The Diesel engine needs no ignition device. In the semi-Diesel engine, however, the compression temperature is not by itself high enough to ignite the oil, which is therefore made to enter a specially hot chamber at the end of the cylinder. Contact with the hot walls of this chamber in the presence of the heated air ignites the charge. This hot chamber is heated by a lamp on starting, but afterwards maintains itself

at a sufficient temperature. The whole of the oil is injected at once and not gradually, as in the Diesel.

It is claimed that the fuel consumption of a Diesel engine need not be more than 0·45 lb. of oil per B.H.P.-hour at full load. If the oil have a calorific power of 15,000,000 ft.-lb. per pound, this is equivalent to the engine yielding 1,980,000 ft.-lb. for every $0{\cdot}45 \times 15{,}000{,}000$ ft.-lb. put into it, giving

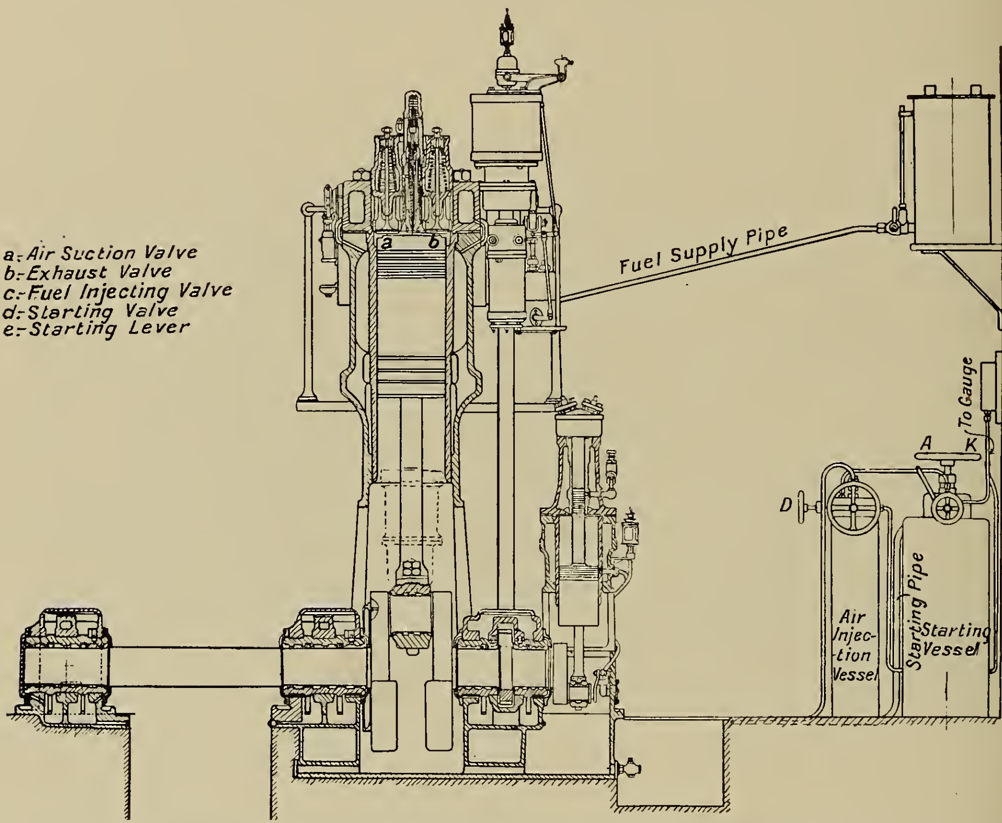


FIG. 70.—Sectional view of Diesel Oil Engine (Mirrlees, Watson & Co.), Note position of inlet valves. *See also* Fig. 70A.

a brake-thermal efficiency of $\frac{1{,}980{,}000}{6{,}750{,}000}$ or nearly 30 per cent. This is a high efficiency, and the reason why it can be obtained is mainly on account of the high compression employed. Reference to p. 85 will show that the "gas standard" efficiency for a similar compression ratio, of about 12, is 52 per cent.

The Diesel engine may be made two-stroke or four-stroke as desired; in the former case it is easily made reversible,

which for marine work is convenient. Any widespread use of this engine either on sea or land depends very largely on a supply of residual oil fuel or of a suitable tar-oil being obtained at a cheap price.

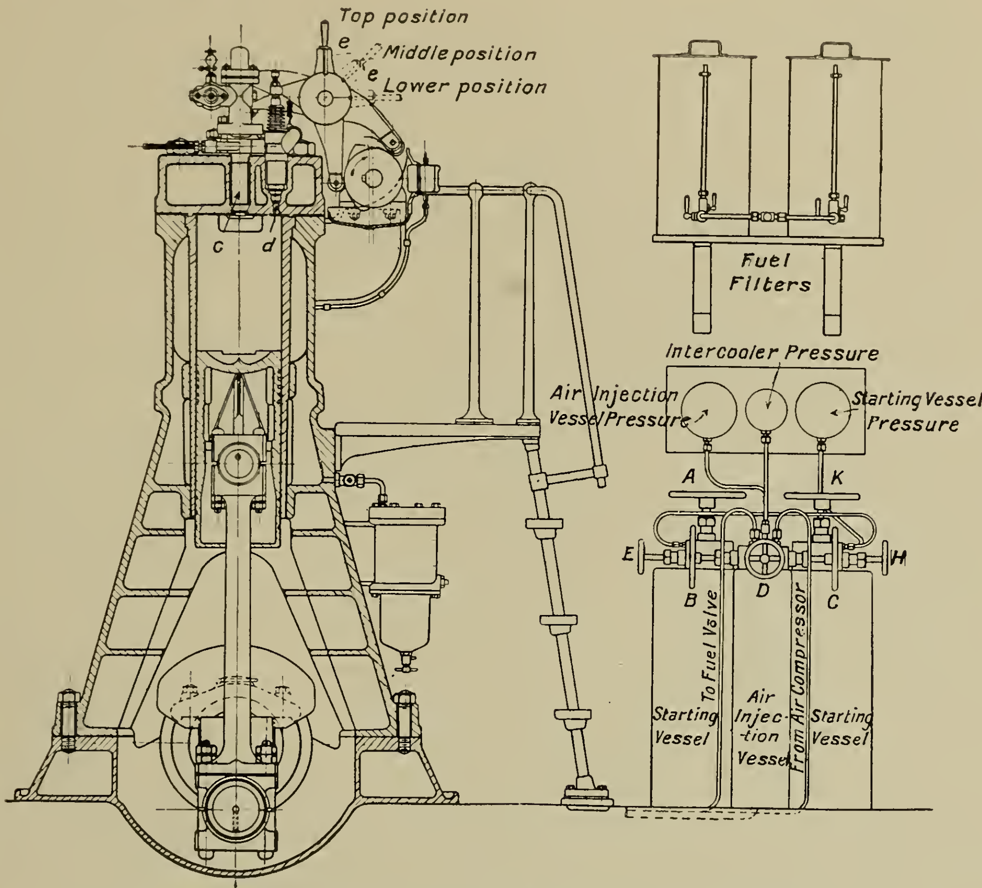


FIG. 70A.—Side view of Engine shown in Fig. 70.

148. The Thornycroft marine engine can be operated with either petrol or paraffin. It is, of course, easier to work a marine engine on paraffin than a land one, as in the former the starting torque required is very slight, and the speed at which the engine runs is much more even. There are, in short, no hills to climb.

The Thornycroft engine is illustrated in Figs. 71 and 72, and the following description will help to elucidate them.

In the first place it will be noticed that the engine is essentially a marine one, the bearing arms being cast on the bottom

half of the bed-plate, and large doors being fitted in the upper half to enable adjustments to be made to the bearings, etc. It will also be noticed that the engine is substantially built and suitable for heavy continuous running at full power.

The cylinders M are cast in pairs with large water-jackets N surrounding them; these water-jackets extend sufficiently far down to enable the working parts of the cylinders to be completely covered. O is the piston fitted with five piston rings; P the connecting rod working on the gudgeon pin Q fitted with a solid bush. R is the crankshaft, and it will be noticed that the cranks are at 180 degrees with each other. The main bearings are shown at S, and are of considerable length. The bottom ends T of the connecting rods are adjustable, and it will be noticed that to assist lubrication the cap and bottom half brasses are left slightly narrower than the top half. The pinion U on the crankshaft drives two fibre wheels V connected to the half-speed shafts. The free-wheel

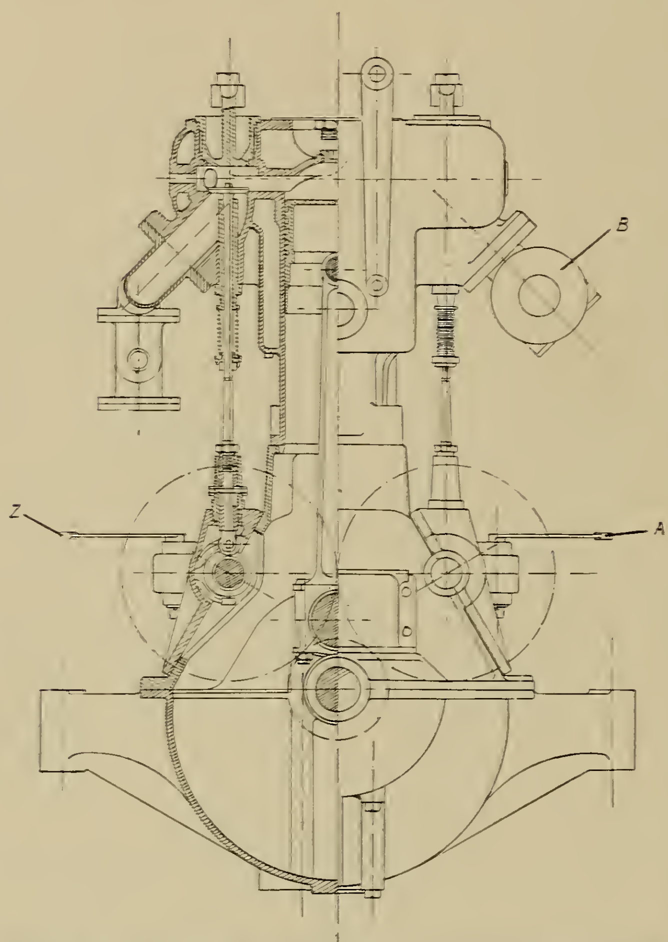


FIG. 71.—General Arrangement of Thornycroft 6″ × 8″ Marine Petrol or Paraffin Engine—End view.

starting arrangement is shown at W, together with the handle and chain wheels. The sparking plugs are shown at XX and are of the positive make-and-break type worked by tappets

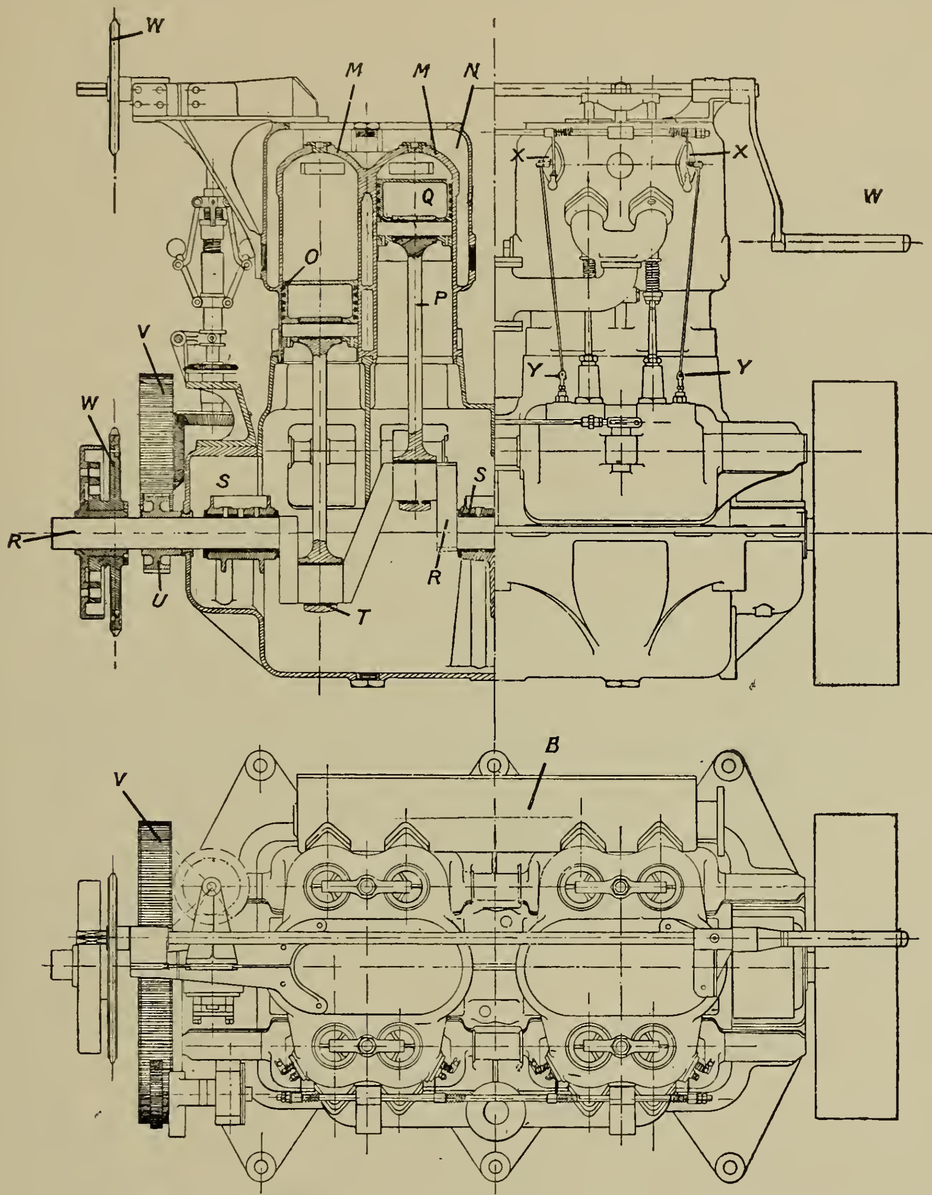


FIG. 72.—General Arrangement of Thornycroft 6″ 8″ Marine Petrol or Paraffin Engine—Side view.

YY. Advance sparking gear is worked by the lever Z, and half compression for starting by the lever A. The exhaust collecting-branch is water-cooled.

149. The Petrol Engine.—The principle of working of a petrol engine is just the same as that of a gas or oil engine—so much so that petrol engines have not infrequently been coupled up to suction producers and run for a time as gas engines. Although this is so it must be borne in mind that owing to differences in the nature of the working fluid the

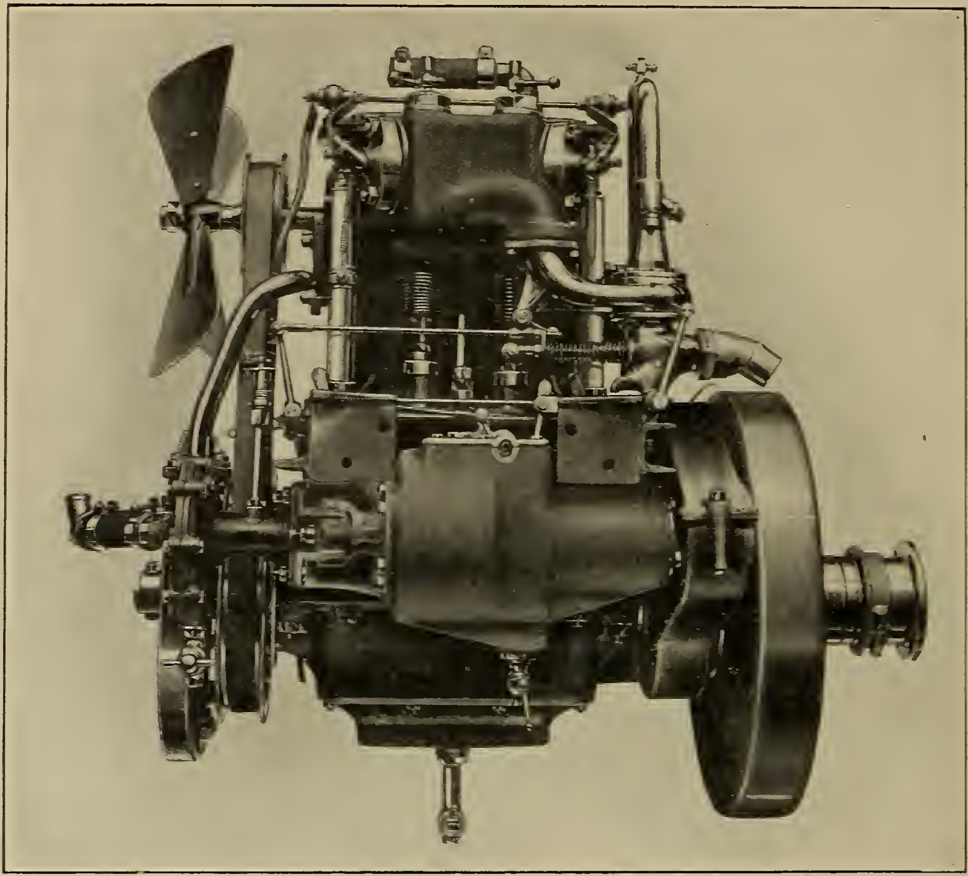


FIG. 73.—16 H.P. Two-Cylinder Albion Petrol Engine for Motor Wagons.

proportions of the engines require to be designed separately for each method of working. In a petrol engine the working fluid is a mixture of air with about 2 per cent., by volume, of petrol vapour. This mixture is formed by admitting both air and petrol to a device called a carburettor. From the carburettor the mixture passes to the engine—most often through a throttle valve of the butterfly wing variety. The proportions of air and petrol are adjusted by having variable

inlets for the air and controlling them by hand or by a governor.

One, two, three, four, six or eight cylinders may be used to make up one engine. The cheaper cars usually have two or four cylinders, whilst six cylinders are often fitted to the larger ones. The more cylinders an engine has the more uniform is the turning moment and the lower the speed at which the engine can be run without stopping. This is an important point, and it is usually discussed under the title of "flexibility." Maximum H.P. is usually obtainable at from 1,500 to 2,000 r.p.m., but it is often desired to run at much lower speeds. As the car speed is required to have a very considerable range, and as full power should be available at low

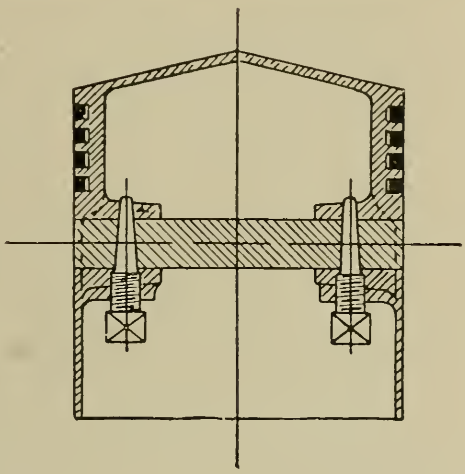


Fig. 74.—Typical Piston of Petro Engine.

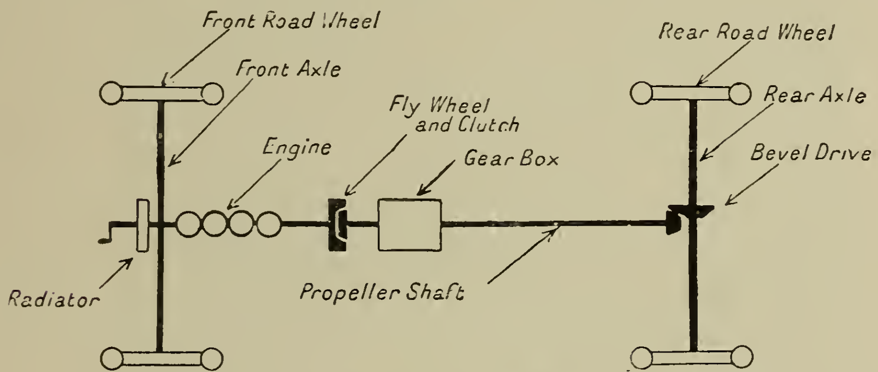


Fig. 75.—Line Diagram of arrangement of Motor Car Chassis. (Compare with Fig. 76.)

as well as at high speeds, variable gearing has to be introduced between the engine and the road wheels.

This brings us to the consideration of the mechanism by which the power of a petrol engine is transmitted to the road wheels of a car. This is shown diagrammatically in Fig. 75, whilst in Fig. 76 is seen a Talbot Chassis showing how the arrangement is carried out in practice.

The engine is fitted to the car so that the crankshaft points in the direction of motion of the car; this shaft is continued

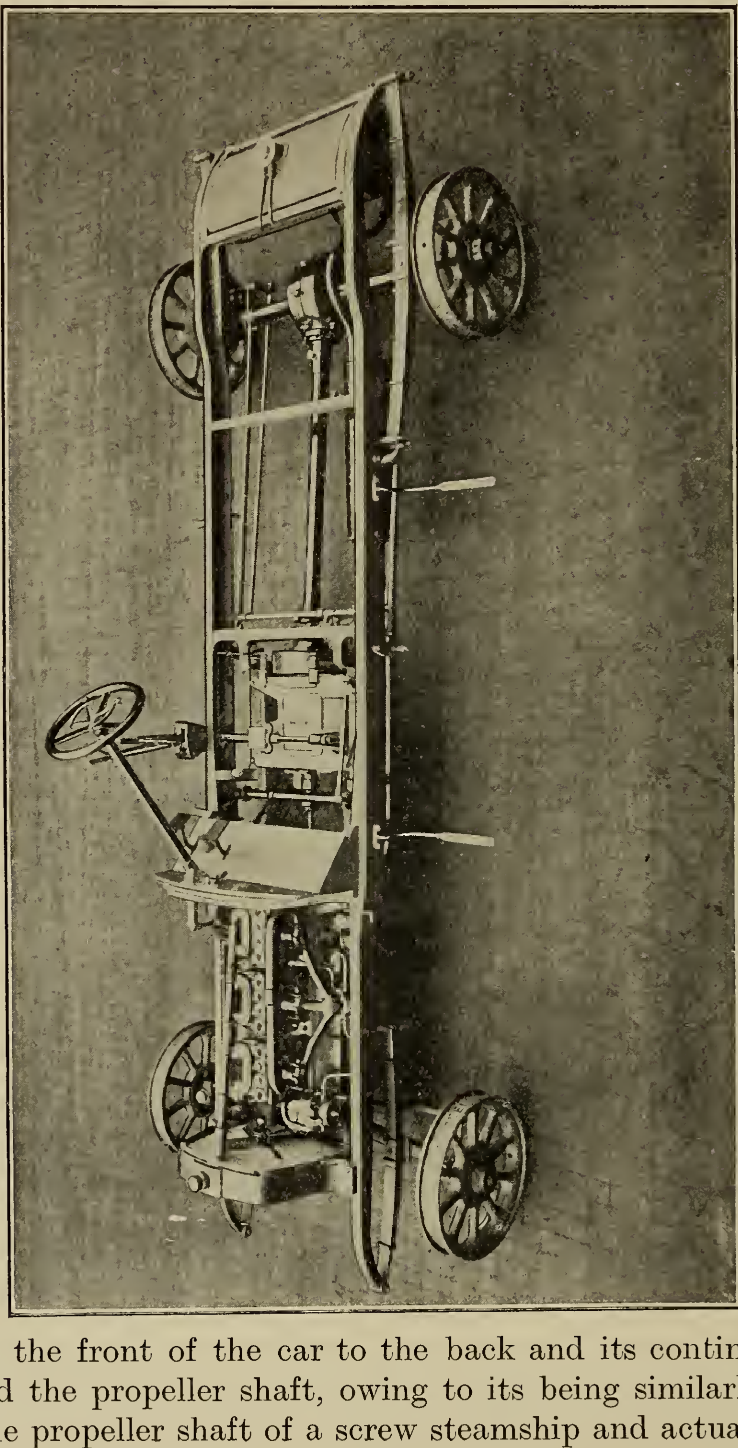


FIG. 76.—Chassis of 20 H.P. Six-Cylinder Talbot Car.

from the front of the car to the back and its continuation is called the propeller shaft, owing to its being similarly placed to the propeller shaft of a screw steamship and actually being

the propeller shaft when used in a marine motor. This shaft transmits power by bevel, worm, or chain gearing to the rear axle of the car, which of course is at right angles to it. In order to be able to alter the velocity-ratio between the engine and rear axle a gear box is fitted in the position shown, and by means of a lever worked by hand, the velocity-ratio can be altered at will.

The **mechanical efficiency** of the transmission from engine cylinder to road wheels is variously stated as anything from 60 to 80 per cent. The following table based on tests by Riedler shows generally the way in which the losses are incurred.

	H.P. at full speed on level			
	Benz Car (84 m.p.h.)	Adler Car (71 m.p.h.)	Daimler-Knight Car (50 m.p.h.)	Büssing Wagon and Trailer (16 m.p.h.)
	H.P.	H.P.	H.P.	H.P.
Loss in transmission .	17	12	6	13
Total rolling loss . .	27	$23\frac{1}{2}$	9	17
Front wheel friction and windage	4	$3\frac{1}{2}$	4	4
Air resistance . . .	52	36	22	4
Total H.P. . .	100	75	41	38

The effect of speed upon the air resistance is very well seen from these figures. It rises from the 4 H.P. of the 16 m.p.h. vehicle to 52 H.P. for the 84 m.p.h. car.

150. Sleeve and Rotary Valves for Petrol Engines.—A few petrol engines are fitted with what are known as "sleeve valves" instead of poppet valves. Sleeve valves work on much the same principle as a steam engine slide valve. The moving sleeve works in between the cylinder liner and the cylinder wall, and the liner itself may be made to slide too. In this way timed openings and closings of valve passages are possible. A further proposal is to use rotary valves, in which, by the steady rotation of one long distributor, all the cylinders are controlled. At present, however, experience is in favour of the poppet type of valve.

151. Aeroplane Engines.—Aeroplane engines, although generally similar in principle to motor-car engines, require to be made much lighter. The seven-cylinder Gnome rotary engine is compact and very light for the horse-power developed, its weight being only about 3 lb. per horse-power. It has been much used for aeroplane work, but owing to the difficulty of silencing the exhaust from the rotating cylinders, and to its

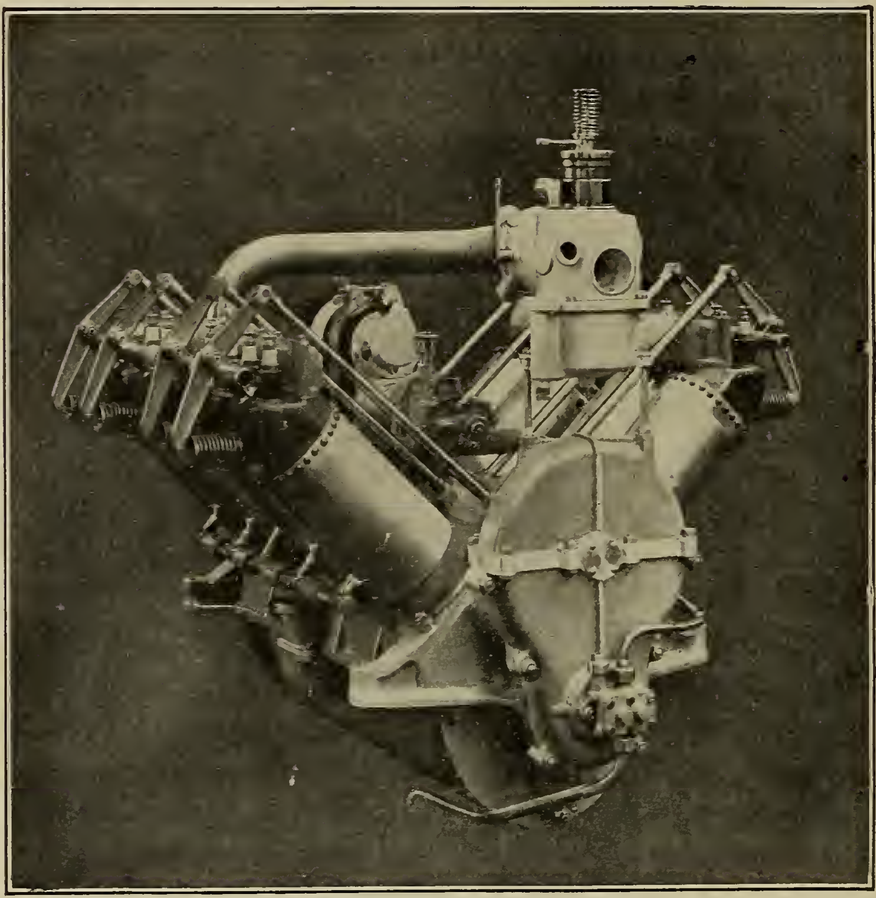


FIG. 77.—Wolseley Aeronautical Engine.

large lubricating oil consumption, it is not so desirable, especially for military purposes,* as a non-rotary engine which, although perhaps heavier, can have a silencer fitted.

In Fig. 77 is shown a small, 60 H.P., non-rotary engine

* At the War Office trials in 1914 the chief prize was awarded to the 100 H.P. water-cooled Green engine. Smaller awards were given for aircraft engines of the following makes—Dudbridge, Wolseley, Beardmore, Gnome, Argyll, British Anzani, Sunbeam.

built by the Wolseley Co. for use in aircraft. The engine has 8 cylinders, each $3\frac{3}{4}$ in. bore × $5\frac{1}{2}$ in. stroke. The cylinders are of the separate type, mounted on an aluminium crank case, and placed at an angle of 90 degrees laterally. The cylinder jackets are of spun aluminium, screwed and jointed to the combustion heads; the induction valves are atmospheric. The pistons are of drawn steel, machined and ground to gauge. The engine is guaranteed to develop continuously not less than 60 B.H.P. at 1,250 r.p.m., with a petrol consumption of not more than 0·7 pints per B.H.P. hour, and a maximum of 68 B.H.P. at 1,400 r.p.m. for a duration of 10 minutes. The total weight of the engine as specified, complete with magneto, wiring, plugs, all water pipes on engine, water pump, oil pumps, piping and connexions, but exclusive of flywheel, exhaust pipes and silencer, does not exceed 300 lb., or 5 lb. per H.P. obtainable.

Particulars * are given on p. 248 of the more important features of the engines exhibited at the 1914 Aero Show at Olympia.

152. Carburettors.—The function of a carburettor is to intermingle the petrol or other fuel with the air, so that an explosive mixture is formed which can be admitted forthwith to the cylinder. It is possible either to allow only a portion of the air to pass through the carburettor and then to add additional air to the mixture so as to bring it to the required proportional composition, or, on the other hand, the whole of the air may be passed through the carburettor. Some petrol comes over in the form of a liquid spray, and air carrying such a spray is quite easily explosive.

The "intermingling" is caused in one of two ways: (1) by the **jet method** (which is the most common), or (2) by the **surface evaporation method.** In the former the jet may be of either of the varieties shown in Figs. 78 and 79. In the former the air sucked through B, on the opening of the valve, causes petrol to rush up the pipe A, past the screw-adjusted inlet opening to a small hole on the conical seating of the valve. The lift of the valve therefore not only admits air but uncovers the small petrol hole up which a jet of petrol at once squirts.

* *The Automobile Engineer*, April 9, 1914.

Aircraft Engines

Nominal H.P.	Name	No. of Cylinders	Bore and Stroke	Type	Capacity	Normal R.P.M.	Petrol Consumption per B.H.P. Hour	Oil Consumption per B.H.P. Hour	Weight all on
			mm.		c.c.		pint.	pint.	lb.
120	Argyll	6	125 × 175	Vertical	12,885	1,200	·55	—	600
90	Austro-Daimler	6	120 × 140	Vertical	9,500	1,300	·6	·025	460
120	Austro-Daimler	6	130 × 175	Vertical	13,937	1,200	·6*	·025*	630
85	Benz	6	106 × 150	Vertical	7,942	1,300	·55	·025	435
100	Benz	6	116 × 160	Vertical	10,146	1,300	·5	·025	500
150	Benz	6	130 × 180	Vertical	14,335	1,300	·5	·025	610
100	Curtiss	8	102 × 127	90° V	8,302	1,800	·58	—	380
60	Green	4	140 × 146	Vertical	8,990	1,200	·53*	·11*	353
100	Green	6	140 × 152	Vertical	14,039	1,200	·64*	·077*	525
80	Gnome single-valve	7	110 × 150	Radial	9,978	1,200	—	—	—
100	Gnome single-valve	9	110 × 150	Radial	12,829	1,200	—	—	—
85	Mercédès	6	105 × 140	Vertical	7,274	1,400	·6	·037	412
105	Mercédès	6	120 × 140	Vertical	9,500	1,350	·6	·037	539
70	Renault	8	96 × 120	90° V	6,949	1,800	·8*	·083*	396*
100	Renault	12	96 × 140	60° V	12,160	1,800	·8	·08	638
90	Salmson	7	120 × 140	Radial	11,084	1,250	·6	·025	405
130	Salmson	9	120 × 140	Radial	14,250	1,250	·6	·025	585
200	Salmson	14	120 × 140	Radial	22,167	1,275*	·6*	·125*	1,045
150	Sunbeam	8	90 × 150	90° V	7,634	2,500	·6	·04	540
225	Sunbeam	12	90 × 150	60° V	11,451	2,500	·6	·04	765
75	Wolseley	8	96 × 140	90° V	8,107	1,800	·62	·007	405
90	Wolseley	8	102 × 140	90° V	9,152	1,800	·62	·007	420
100	Wolseley	8	96 × 140	90° V	8,107	1,800	·6	·005	475
130	Wolseley	8	127 × 178	90° V	18,039	1,200	·6	·005	785

Nominal H.P. and normal revolutions as stated by makers. The weights given include radiator and water unless engine is air-cooled.
* Denotes that the figures have been procured from independent tests; consumption figures otherwise are those given by the makers.

Then the petrol, having so low a vaporizing point, at once turns into vapour and forms with the air an explosive mixture. The screw adjustment—or needle valve as it is called—allows the richness of the mixture to be adjusted. Fig. 79 shows a better and more familiar way of doing the same thing. Air is sucked in past the nozzle of the jet F and out by the opening K. In rushing past the jet it sucks up a petrol spray, which evaporates as it mixes with the air.

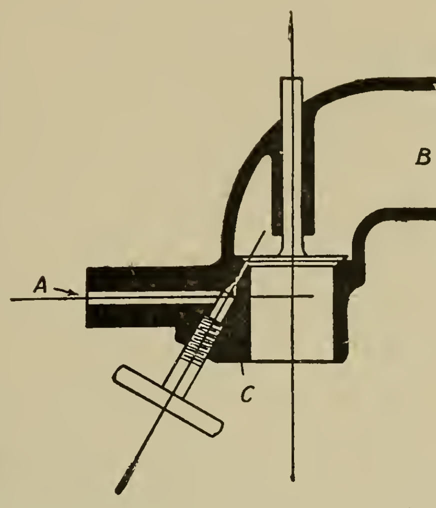


FIG. 78.—*A*. Petrol inlet to Controller. *B*. Inlet pipe to Engine. *C*. Needle valve to regulate flow of petrol.

On the left of the figure is seen the float chamber for keeping the petrol level constant. It operates much as does a ball and cock feed to a water cistern. B is a needle valve which gets pushed down on to its seating by the levers D when the float C rises to the top. This stops more petrol coming in until the petrol-level sinks so much as to let the float down till the levers open the needle valve again, when more petrol flows in. The weight of the float is so adjusted that the petrol-level is kept at just the right height. It is the custom to have the petrol standing just below the top of the jet, but it works even if standing much below the top of the jet. Evidence of

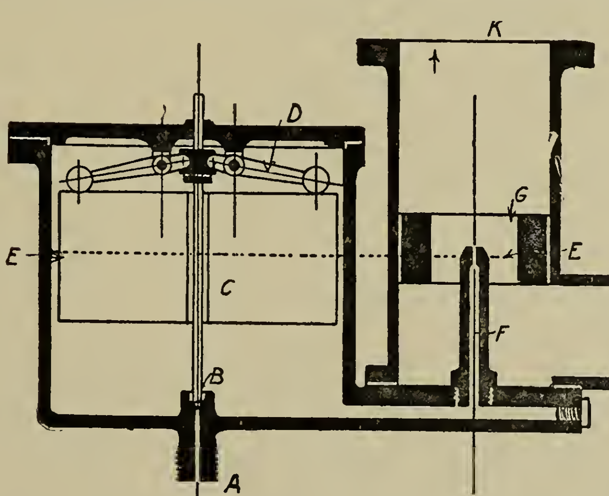


FIG. 79.—Jet Carburettor and Float Chamber. *A*. Petrol inlet. *B*. Needle valve. *C*. Float which closes the needle valve *B* through the levers *D* when the petrol reaches the level *EE*. *F*. Petrol jet. *G*. Air nozzle. *K*. Inlet pipe to engine.

this is seen in instances in which the float chamber is sucked quite dry during the running of the engine when the petrol inlet pipe A has got choked up in some way. The principle of the working of the jet will be gone into later. An efficient type of a surface carburettor is shown in Fig. 80, which illustrates a carburettor used on a Lanchester engine. The principle of its working is obvious from the diagram. The air passing over a large petrol-soaked surface takes up petrol vapour.

All the carburettors described work best with a certain

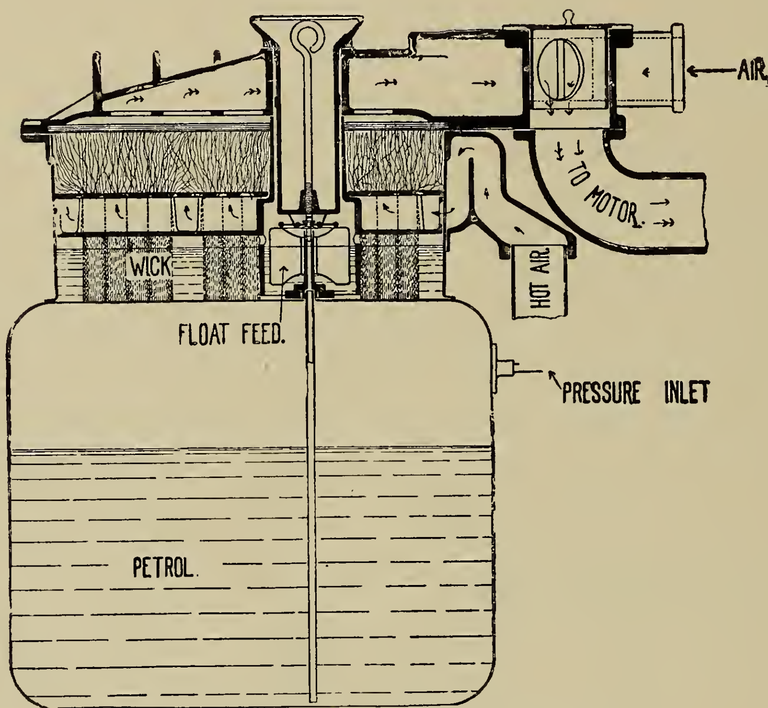


Fig. 80.—Lanchester Surface Carburettor.

velocity of flow of the air. When, however, the engine runs very fast or very slow the air velocity changes accordingly so that the carburettor sometimes gets more air, and sometimes less, than it wants. If the flow of air be increased it is found that too much petrol is taken up, so it is customary to arrange for only part of the air to pass the jet and for the rest to be added to the mixture without passing the jet at all. It is best to arrange for this adjustment to be made auto-

matically, and Figs. 81 and 82 show how this can be done. The former shows the Krebs automatic carburettor. When, owing to increase of piston speed, the suction on the air increases, the leather diaphragm O is sucked down against the weak spring P and opens a valve at M so that air can flow in and

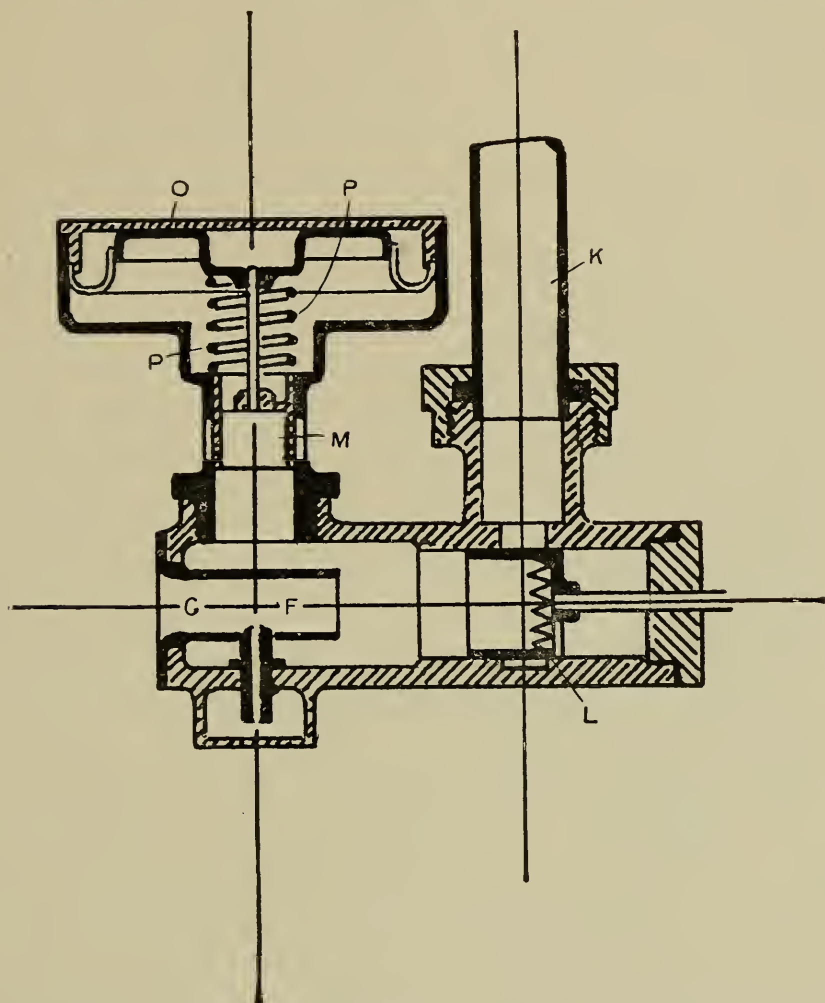


FIG. 81.—Krebs Carburettor, in which the opening to the extra air supply is controlled by the suction of the engine.

mingle with the air which has entered at G and has passed the jet F. L is the throttle valve controlling the quantity of the mixture which is allowed to pass to the cylinder by the pipe K. Fig. 82 shows another way of doing the same thing. As the suction increases the extra air comes in through the valve M and joins at K the part which has come in over the jet F.

There are many other ways, easily devised, of applying the same principle.

Owing to the heat absorbed by the evaporation of the petrol it is usual to warm the entering air slightly. This is done by putting the air inlet pipe nozzle close up to one of the exhaust pipes so that the air in rushing past the hot pipe gets warmed slightly. Of course the fact that the whole of the carburettor is under the warm engine bonnet helps to keep the temperature from falling too low.

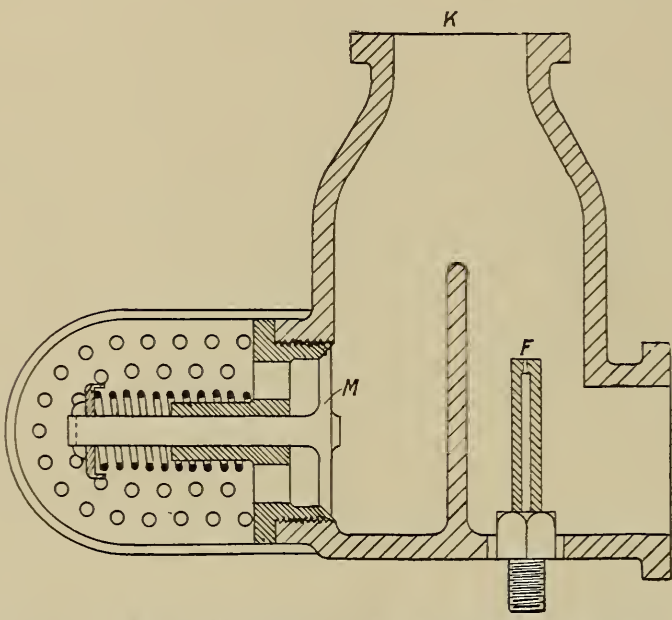


Fig. 82.—Automatic Carburettor working in a generally similar way to the Krebs. The opening of the valve M depends on the suction.

When paraffin is used as a fuel much more heat is necessary.*

153. Special Forms of Carburettor.—There are a great number of special forms of carburettor, for which the inventors claim remarkable advantages. Of these the best known are the White and Poppe, the Claudel-Hobson, and the Zenith, but there are many others in widespread use. Having regard to the great variations in the manner of working of these carburettors, it is surprising that so many of them work so uniformly well. The White and Poppe is illustrated in Fig. 83. It is probably the most successful of them all in respect of fuel economy. It has no additional air inlet, and keeps the mixture correct by manipulating the jet and the air throttle. The jet is placed in the axis of a cylindrical chamber across which the air flow is directed. This chamber is enclosed in a

* *Vide* Par: 141.

metal sleeve, and the whole has a circular airway drilled through both sides. As the chamber is caused to rotate slightly, the air passage is restricted. This restriction is made also to affect the jet owing to the petrol passage-way up the jet being drilled a little eccentrically to the axis of the jet. A cap similarly drilled fits over the jet, and as the jet-cap and the chamber rotate through an angle, the effective jet opening is decreased in the same proportion as in the throttling of the airway. This proportion is obtained by the fact that in each case it is a circle sliding over a circle, and that both are fully opened and fully closed, together. The illustration shows a double sleeve, the parts of which can be set at a fixed relationship to one another as a means of adjustment.

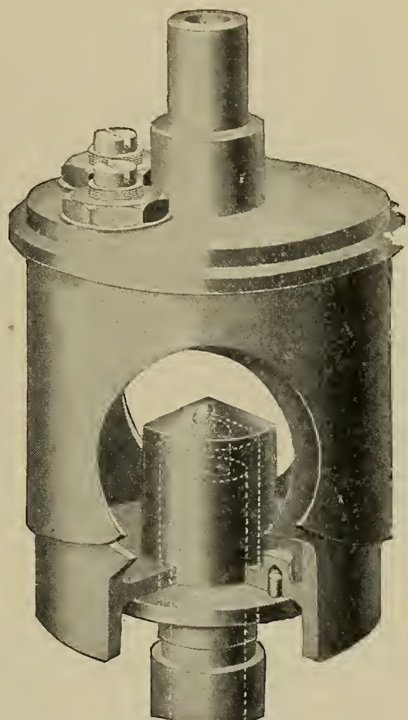


FIG. 83.—Interior of White and Poppe Carburettor.

154. Heating Air Supply to Carburettor.—The various liquid fuels used in internal combustion engines require different methods of carburation.* With light spirits like petrol it is only necessary to warm the air which passes over the jet, or to jacket the carburettor with the warm water which circulates through the radiator. But with a fuel like paraffin it is necessary to heat either the air or else the mixture very considerably. The former is sometimes done by passing the air through tubes heated to a high temperature by the hot exhaust gases circulating around them. This hot air is then passed over the paraffin jet, and the paraffin is carried along partly as spray and partly as vapour. This mixture is then too hot to enter the cylinder, and has to be cooled by mixture with some cold air. But even so it has to enter the cylinder

* *See* Par. 141.

hotter than is customary with petrol, and a less weighty charge is therefore employed which leads to something like 10 or 15 per cent. less horse-power being developed. When paraffin is used the compression pressure needs to be lowered to about 65 lb. per sq. in. instead of the figure of 80 lb. per sq. in., or more, which is usual for petrol (both gauge pressures). As was stated in par. 135, paraffin is less homogeneous than petrol, and there is usually some carbon deposited; the cylinders therefore need more frequent cleaning. The presence of this deposit tends to induce pre-ignition, and for that reason the compression ratio is kept low.

155. The Cottrell Paraffin Carburettor.—This carburettor is illustrated diagrammatically in Fig. 84. C is the pipe which receives the air and paraffin spray coming over from the jet in the carburettor marked H. When the fuel gets to the branch pipe it divides right and left to either end of the vaporizer M. M is shown in section at the lower right-hand side of the figure. It consists of a corrugated pipe which is surrounded by hot exhaust gases and conveys in its interior the air and paraffin mixture. This corrugated pipe has to be kept hot. In starting the engine cold, provision is made for working on petrol for two or three minutes and then, the pipes M having got hot, the paraffin is turned on. At B there passes a mixture of heated air and paraffin vapour. The air is much less in proportion than would ignite, so at F an adjustable inlet is fixed to admit more air until the mixture is of the correct proportions. The purpose in not letting all this air in earlier is that with a less proportion of air the paraffin particles get more effectively heated and the velocity of passage through the vaporizing tube is slower.

A variation of this arrangement has been tried for use in the tropics, whereby air only was passed through the star tubes M, then through a lagged pipe to an ordinary jet chamber. This alternative arrangement has worked well, but in temperate climates hardly seems so effective as the unmodified type of carburettor. To begin with, the paraffin draws all its heat from the air which sweeps it along, instead of by actual rushing contact with the hot vaporizer tubes. Further, the mixture enters the cylinder much sooner after its creation

than in the other arrangement, and it is evidently advantageous to allow time for the paraffin vapour and air to mix intimately with each other. A drop in horse-power in the engine must be expected of 10 to 15 per cent. compared with the horse-power obtainable when using petrol. This loss is chiefly due

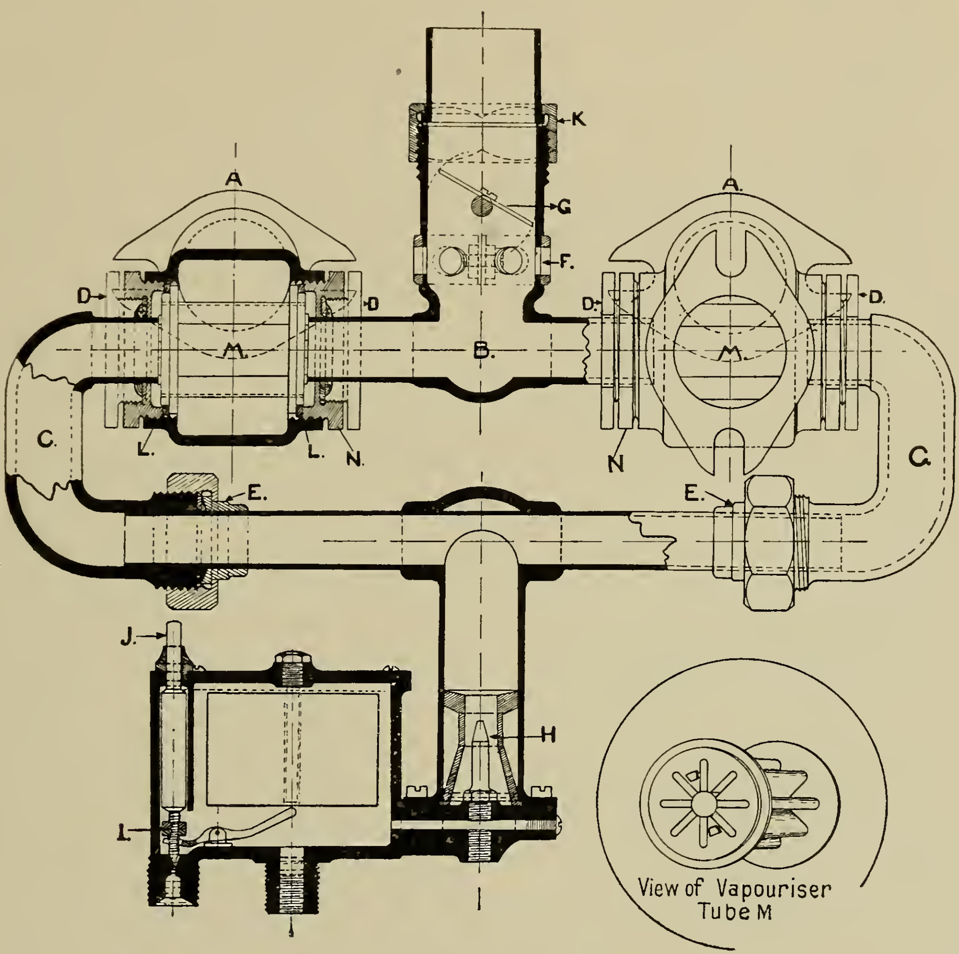


FIG. 84.—Cottrell Paraffin Carburettor. Shown diagrammatically.

to the higher temperature of the entering charge which for a given cylinder volume naturally reduces the weight admitted. There is also a loss owing to the necessary lowering of the compression ratio and consequent lowering of thermal efficiency. It has sometimes been thought that this lowering of the compression is due to fear of the charge being pre-ignited

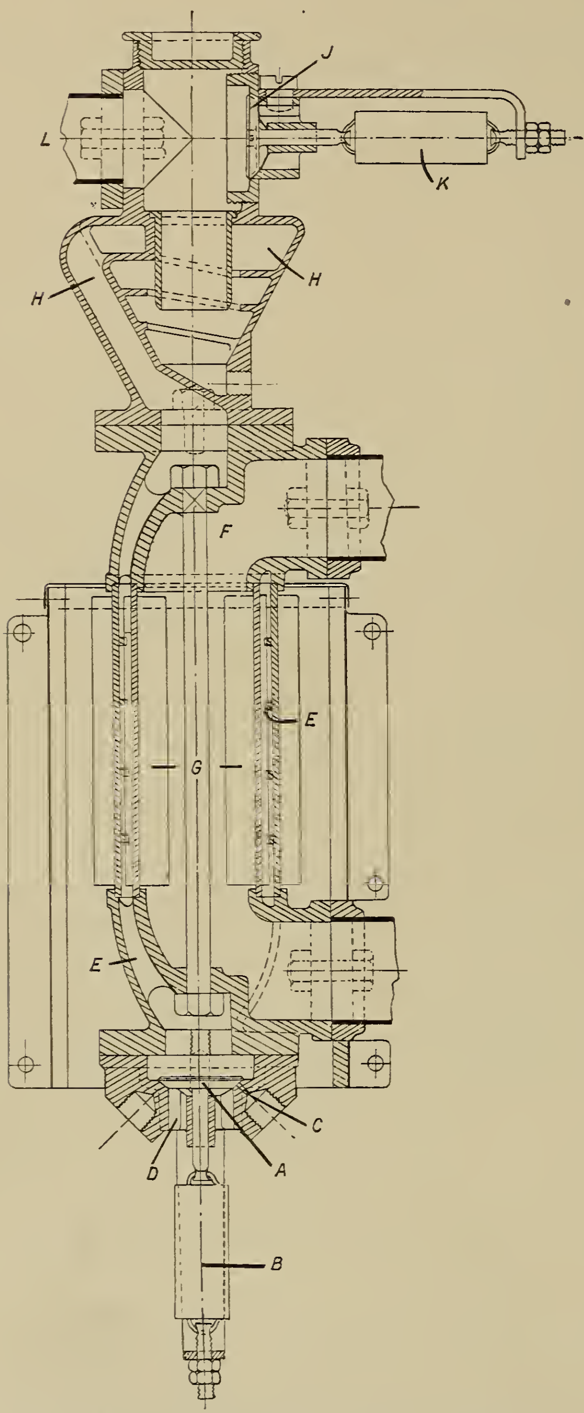


FIG. 85.—Thornycroft Paraffin Carburettor.

through the low temperature at which paraffin vapour and air ignite spontaneously; but the real reason is that given in the previous paragraph.

There is a gain, however, in that the calorific value of paraffin per pound is about 18,000,000 ft.-lb. against about 15,000,000 for petrol; also an advantage is found in the smaller consumption of lubricating oil owing to the lubricating properties of the paraffin itself.

156. The Thornycroft type of paraffin carburettor is illustrated in Fig. 85. Its manner of working is generally similar to that of the Cottrell, but the heating surface is less in proportion. It works well in practice, and its mode of operation is as follows :—

The oil is drawn into the vaporizer together with a certain amount of air by the suction of the engine; this mixture is then passed through a tube which is kept heated to a fairly

high temperature by the exhaust gases coming from the engine. This thoroughly vaporizes and intimately mixes the vaporized oil and air. The mixture is then passed through a spiral separator, which separates any solid matter from the vapour, is mixed with extra air as required to form an explosive mixture, and then passed through the throttle to the cylinders. In the drawing, A is the inlet valve for both oil and air, the valve being under the action of spring B, which normally keeps the valve closed and the oil supply C shut off; the oil enters by holes in the seating. The mixture then passes along the annular space EE which is kept heated to a high temperature by the exhaust flowing through the centre of this annular chamber as shown at F. The annular chamber, it will be noticed, is fitted with gills G to enable a maximum amount of heat to be supplied to the mixture. H illustrates the separator for removing the solid particles from the mixture, and J the "extra-air" inlet, under the control of the spring K. The tension of this spring and also of that governing the inlet of the mixture to the vaporizer can be varied by a screw and nut as shown. This adjustment is made when the engine is on the test-bed before the brake trials. The outlet from the vaporizer to the motor is by pipe L. There are numbers of other paraffin carburettors, but the principle of operation is generally similar to those above described.

157. Theory of Jet Carburettors.—The energy equation for the flow of any fluid (liquid or gas) is as follows—*see* Perry's *Applied Mechanics* :—

$$\frac{v^2}{2g}+\int\frac{dp}{w}+h=\text{constant.} \quad . \quad . \quad . \quad . \quad (1)$$

This is true for any stream line.

v = velocity of fluid in ft. per sec.
g = 32·2.
p = pressure in lb. per sq. ft.
w = weight in lb. of one cu. ft.
h = height in feet above datum.

If this equation be applied to the flow of air through the carburettor due to the suck of the engine, it may be simplified

in many ways. We want to find the amount by which the pressure in the rushing air is made lower than the atmospheric pressure owing to the suck of the engine pistons. We start with air at atmospheric pressure p_o, and density w_o, at no velocity, and we end with air at velocity v_0, pressure p, and density w. It can then be shown from equation (1) and the adiabatic relation for air that :—

$$\frac{v_o^2}{2g} = \frac{\gamma}{\gamma - 1} c^{\frac{1}{\gamma}} \left\{ p_0^{\frac{\gamma-1}{\gamma}} - p^{\frac{\gamma-1}{\gamma}} \right\} \quad . \quad . \quad . \quad . \quad (2)$$

It is, however, possible to use a simpler expression than this. Equation (2) assumes that there is no heat lost or gained as the air passes through the carburettor, which is not strictly true. Since some degree of approximation is necessary for simplicity we may without any more serious inaccuracy neglect the change in density of the air, and it then follows from equation (1) that :—

$$\delta p = \frac{w_0 \cdot v_0^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The pressure at the top of the petrol jet is therefore lower than the pressure on the surface of petrol in the float chamber by δp where δp is given by equation (3). A similar argument* to the above will show that the flow of the petrol is determined by a formula of the same type. For a liquid such as petrol in which w is independent of p, equation (1) becomes

$$\frac{v^2}{2g} + \frac{p}{w} + h = \text{constant}.$$

Applying this equation to the stream line which begins at the free petrol surface in the float chamber and ends in the jet spray, and assuming that the top of the metal jet is a height h above the petrol level (so that petrol has to rise through a height h in the jet before it gets to the actual nozzle) we have

$$o + \frac{\delta p}{w} + o = \frac{v^2}{2g} + o + h$$

where v = velocity of flow of petrol and w = density of petrol (i.e. the weight per cubic ft.).

* Neglecting viscosity.

Therefore $$\frac{v^2}{2g}=\frac{\delta p}{w}-h$$

Now by equation (3) :— $$\delta p=\frac{w_0 v_0^2}{2g}$$

so that $$\frac{v^2}{2g}=\frac{w_0 v_0^2}{2wg}-h$$

or $$v^2=\frac{w_0}{w}v_0^2-2gh \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This shows that v_0, the velocity of air flow, must have the value $\sqrt{2gh\frac{w}{w_0}}$ before any petrol will flow at all. It is a matter of common experience that if the rate of air flow doubles the petrol flow more than doubles. Let us see if this is so according to this formula.

First, let $v_0 = a$, and then equal $2a$. The ratio of the squares of the petrol flow in the two cases should therefore, according to experience, be more than 4. By formula (4) it is

$$=\frac{\frac{w_0}{w}4a^2-2gh}{\frac{w_0}{w}a^2-2gh}$$

$$=4+\frac{6gh}{\frac{w_0}{w}a^2-2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

It will be interesting to get some quantitative figures for this. What, for instance. will be the ratio if $h = 0{\cdot}04$ feet (or $\frac{1}{2}$ inch) and a is 5,000 ft. per minute (83·3 ft. per sec.) ?

Petrol has a density of about 0·72 so that 1 cu. ft. will weigh $0{\cdot}72 \times 62{\cdot}3 = 45$ lb. Whereas 1 cu. ft. of air weighs 0·075 lb. at atmospheric temperature and pressure.

According, therefore, to equation (5) the square of the ratio of the two petrol velocities will be

$$4+\frac{6\times 32\cdot 2\times 0\cdot 04}{\frac{0\cdot 075}{45}a^2-(64\cdot 4\times 0\cdot 04)}$$

$$=4+\frac{7\cdot 72}{\frac{a^2}{600}-2\cdot 57}$$

The critical velocity is clearly

$$=\sqrt{2gh\frac{w}{w_0}}=\sqrt{2\times 32\cdot 2\times 0\cdot 04\times 600}=\sqrt{1550}=39\cdot 3 \text{ ft./sec.}$$

$=2,360$ ft. per min. Until, therefore, the air had this velocity no petrol would be carried along. If $a=\frac{5000}{60}$ the equation (5) becomes

$$4+\frac{7\cdot 72}{11\cdot 6-2\cdot 57}=4+\frac{7\cdot 72}{9\cdot 0}$$
$$=4\cdot 86.$$

So that when air velocity increases from 5,000 to 10,000 ft. per min. the petrol velocity is 2·2 times as much and the mixture therefore 1·1 times as rich or 10 per cent. richer. This means that the amount of petrol present per cubic foot of air increases by one-tenth part. This inequality is most marked when the velocity of the air is only a little more than what is necessary to feed the petrol, thus if the air velocity be increased from 2,500 ft./min. to 5,000 ft./min. the petrol sucked along would be increased by no less than 430 per cent., i.e. the quantity of petrol would be 5·3 times as great, giving a richness of mixture 2·6 times or more than double what it was before. In this case, therefore, about twice the quantity of air would be needed, i.e. as much again must pass the additional air inlet as already passes the jet. This simple theory accounts for part of the extra air needed. It does not, however, take account of the effect of eddies, that may circulate around the jet at high air velocities, nor does it take viscosity into account.

It is possible that there are still further reasons why extra air is needed when the engine speed increases, but the above are certainly some of them. The air velocity is highest when the engine is running fast and the throttle wide open. When

the engine is running fast but is much throttled, as in a car running very fast down a slope, the vacuum behind the piston never gets filled up with air and the velocity of air past the jet is not therefore very great; going up hill, however, on low gear the engine speed is high and the throttle wide open. So that the velocity of air past the jet is not solely dependent on the engine speed. This makes control of the additional air inlet by the centrifugally operated governor not as uniformly good as could be wished—it only approximates to extreme cases, fitting accurately the average only. When, however, as in the Krebs carburettor, for instance, the opening of the extra air inlet is controlled by the suction, a much more constant mixture is obtained. A carburettor is usually set so that the right mixture comes away from the jet at low speeds with the extra air inlet closed. Then as the speed rises additional air is allowed to pass in. From the preceding calculations it will be clear that one cause of the lack of proportionality between the air and petrol velocities is the fact that the petrol cannot be allowed to stand at a level equal to the height of the jet, as if this condition were arrived at too closely there would be a risk of the petrol overflowing when standing on grades. A further reason which makes it necessary not to run the adjustment too finely is that the level of petrol in the float chamber is bound to vary somewhat not only with the inclination of the float chamber but also with temperature and quality of supply. In order to keep on the safe side the petrol level must be several millimetres below the top of the jet.

158. Ignition.—The oldest form of ignition was to ignite the explosive mixture by a naked flame, which was put into communication with the cylinder through the medium of a sort of slide valve. It is quite obsolete now, but those interested in the history of the subject will find a full account in Dugald Clerk's book.

A later and more successful form was **Tube ignition,** which consisted in having a short vertical tube, in communication with the cylinder end, heated externally by some means. After the engine had been running a short while the lamp could be removed (or the gas jet turned out) and the heat of explosion was enough to keep the temperature up to the requisite point.

It is illustrated in Fig. 86. The explosive charge was compressed by the inward movement of the piston, and a part of it passed into the ignition tube, the temperature of which raised the temperature of the gas to the ignition point. This type of ignition was very largely used for gas engines, but it is now universally displaced by some form of electric ignition. It was also the first method employed on motor cars, for which use it was found to be unsuitable. It is still in use with some oil engines.

Another method of ignition often used with oil engines is to feed the fuel into a hot combustion chamber connected to the cylinder. This method has already been described in par. 146. It works well, and even residual oils can be vaporized and ignited in this way. The method employed on the Diesel engine is of this type.

159. The chief method of ignition is the electric, and it bids

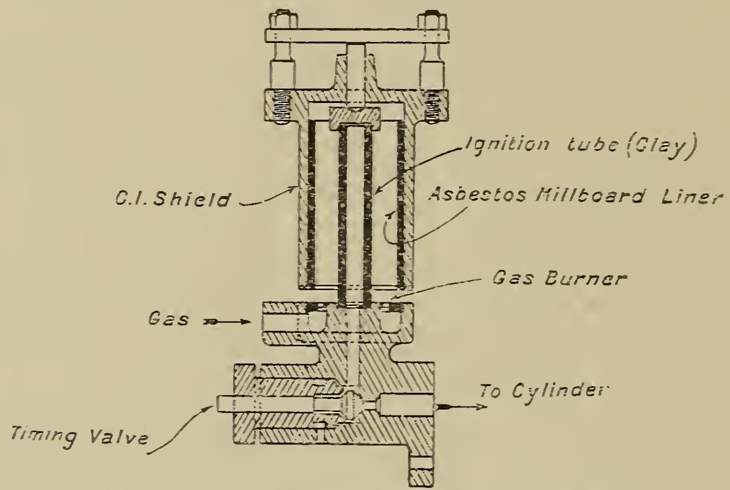


FIG. 86.—Ignition Tube and Timing Valve.

fair to supersede all the others. It is almost equally suitable for gas, oil or petrol engines.

Electric ignition can be carried out by either (1) **high-tension** currents or (2) **low-tension** currents.

The high-tension currents may be obtained in one of three ways: either (1) by batteries or cells furnishing current to an induction coil, or (2) by a small magneto-electric machine (called " magneto " for short) furnishing currents to an induc-

tion coil, or (3) by a magneto furnishing high-tension currents direct to the sparking plug. The low-tension currents are produced from a low-tension magneto. There are therefore many varieties of electric ignition and they may conveniently be set out thus—

Electric Ignition

- High tension
  - Batteries or cells with induction coil
    - Coils fitted with tremblers (1)
    - Coils fitted with fixed " make-and-break " (2)
  - Magneto with induction coil (3)
  - High-tension magneto (4)
- Low tension
  - Low-tension magneto (5)

The oldest is (1) and it is still seen fitted to some petrol engines, although (2) is more common; (3) is relatively rare but was seen in the early Eisemann system; (5) is common practice for engines whose speed is below 300 r.p.m. Method (4) is growing in popularity, and it has the advantage of being simpler to apply to the engine than (5).

Before describing methods (1), (2) or (3) it will be necessary to say something about the induction coil. To those versed in electrical matters it is enough to describe it as a transformer having a straight iron core and a high ratio of transformation.

160. Induction Coil.—The induction coil consists of a soft iron core generally consisting of a bundle of straight iron wires, and on it is wrapped a layer or two of thick insulated copper conducting wire of the primary—or low-tension—circuit. Over this are wound many thousand turns of fine insulated copper wire constituting the secondary—or high-tension—circuit. About 4 volts are applied to the primary circuit and the current repeatedly broken and remade by means of the magnetism of the iron core attracting a small piece of iron mounted on a spring which carries the current. As the spring is attracted inwards it loses contact with a platinum point and so breaks the current. (To make the break the more

sudden it is usual to put a condenser in parallel in the circuit.) This sudden rise and fall of current in the primary causes oscillatory currents in the secondary of a voltage which is higher than that in the primary in the ratio of the number of coils of wire in one to the number in the other. Owing to the effect of the magnetism in the iron core the current in the primary does not rise suddenly to its full value. It follows in fact the law

$$I=\frac{V}{R}\left(1-\epsilon^{-\frac{R}{L}t}\right)$$

where

I = current in amperes.
V = voltage in volts.
R = resistance in ohms.
L = self-induction in henries.
t = time in seconds.
ϵ = base of Naperian logarithms or 2·7183.

The unit of self-induction is the henry. If S be the rate at which the current changes in amperes per second, the back electromotive force produced = L × S. One henry is also defined to be the self-induction of a coil in which, if the current increase at the rate of one ampere per second, the back E.M.F. produced is exactly one volt.

As an illustration of the effect of the above law of rise of the current, take the case of a coil in which R = 500; L = 5·5 and V = 50,000. Then the final and steady value of the current is clearly $\frac{50{,}000}{500}$ or 100 amperes. This current grows up from zero and it is of interest to calculate how long it will take 90 amperes to be the current flowing.

$$90=100\left(1-\epsilon^{-\frac{t}{0{\cdot}011}}\right)$$

or

$$\epsilon^{-\frac{t}{0{\cdot}011}}=1-0{\cdot}9=0{\cdot}10$$

so that t=about $\frac{1}{40}$ second.

The current therefore rises by no means *instantaneously* and this leads to the "make" of the primary current, pro-

ducing a much less vigorous spark in the secondary than does the "break." The break is almost instantaneous; the only thing that tends to prevent it being so, is the energy of rush of the primary current which jumps over the gap in its earlier stages. The energy stored up in the flowing current is equal to $\frac{1}{2}LI^2$, and it is to provide a convenient swamp to absorb this suddenly released energy that the condenser is provided.

161. High-Tension Coil Ignition.—The induction coil is supplied with a low-tension current obtained from either batteries, accumulator cells or a suitable magneto. In any case the principle of working is the same. The spark gap (*see* Fig. 90) is placed in the cylinder as shown in Figs. 87, 88, 89, and 91. A rotating contact kept at a speed proportional to that of the engine and called a distributor distributes the current

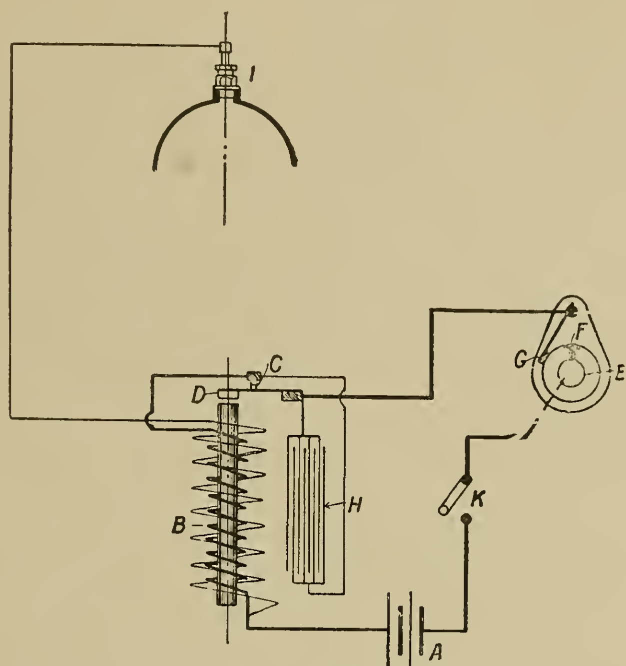


FIG. 87.—Diagram showing mode of working of high-tension ignition with coil and accumulator. *A*, Accumulator. *B*, Induction coil. *C*, Contact breaker. *D*, Trembler. *E*, Commutator on end of cam shaft, for closing circuit at right moment by bringing metal segment *F* against the brush *G*. *H*, Condenser, to make break of current sudden. *I*, Ignition plug in cylinder. The other end of the secondary winding is earthed.

to each cylinder just as it is needed. What happens therefore is this. The trembling blade on the coil—called the trembler—vibrates very rapidly and produces a shower of sparks in the secondary (one spark corresponding to each break of current in the primary). During each contact about a dozen sparks or more may pass. One good spark would be enough and therefore a modification of this method is sometimes em-

ployed. Instead of a trembler actuated by the magnetism of the iron core of the coil, the current in the primary circuit is

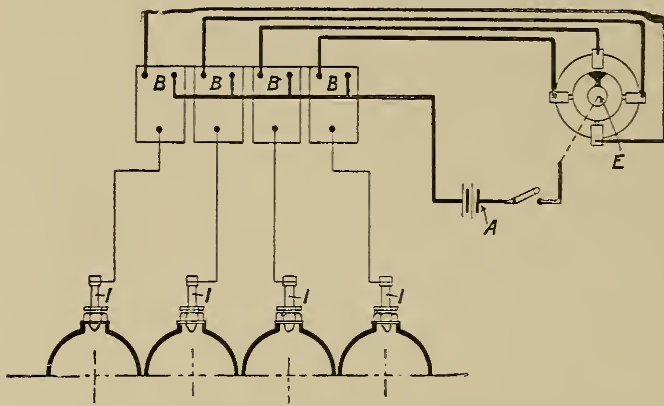


FIG. 88.—Coil and accumulator Ignition, for a four-cylinder engine, with separate coils for each cylinder. *A*, Accumulator. *B*, Coils each with its own condenser and contact maker. *E*, Commutator for distributing current to the various cylinders at the right moment. *I*, sparking plugs.

made and broken by the action of the engine. A mechanical make-and-break is fitted to the half-speed shaft of the engine

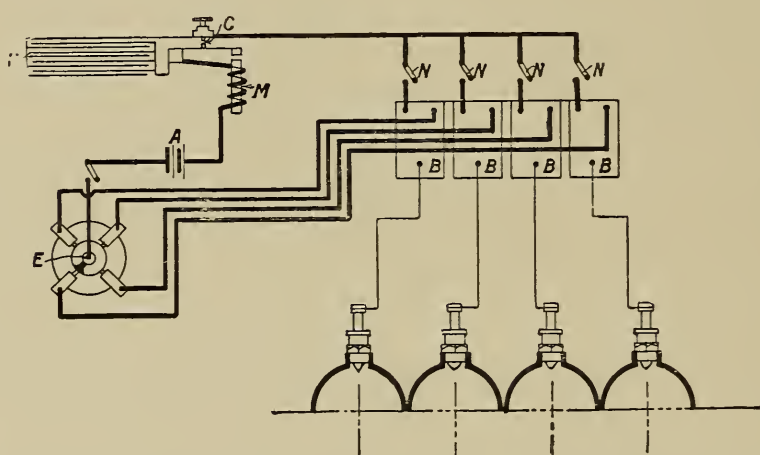


FIG. 89.—The arrangement shown in Fig. 97, except that one trembler serves all the coils. This saves having to adjust each trembler until all are working at same frequency. *C* is the common contact maker, and *N* are switches for cutting out coils when necessary.

so as to produce one spark only in the cylinder. It is possible to ring the changes on this form of ignition so as to produce a great many varieties, although the differences between them

are hardly fundamental. Illustrations, reproduced from Mr. Strickland's useful book, are shown of several such methods (*see* 87, 88, 89, and 91), and the letterpress at the foot of each will suffice to show their differences.

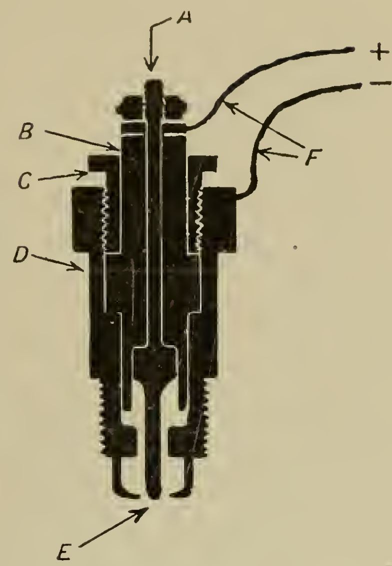


FIG. 90.—High Tension Sparking Plug. *A*, Metal Rod; *B*, Porcelain Insulating Sleeve; *C*, Gland; *D*, Body of Plug screwing into cylinder; *E*, Sparking Points; *F*, Electric Wires from H.T. Magneto or Coil.

162. The **Lodge** (Sir Oliver Lodge) system of ignition is just the ordinary coil and accumulator ignition in which the high-tension current instead of being passed direct to the sparking plugs is made to charge up the inner coatings of two Leyden jars. When the jars are "full" an external spark gap placed in parallel with the jars breaks down and a spark passes. This sudden release of the electric charges on the inner coats of the Leyden jar causes such a rush of current from the outer coating of one Leyden jar to the other, and such a violent oscillation to and fro of the current afterwards that nothing will stand in its path. It breaks through oil films, soot, deposit of all kinds, water or anything else that there may be on the ignition points; owing to its high frequency it also tends to take straight direct courses, and there is little disposition on its part to seek any short circuit

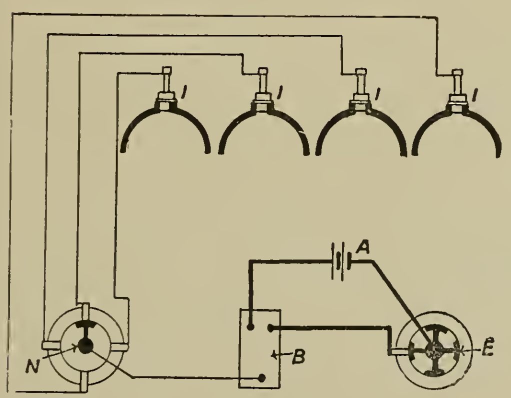


FIG. 91.—The arrangement of Figs. 88 and 89, except that the secondary current is distributed directly, so enabling only one coil to be used for all four cylinders. The disadvantage is that the insulation is more difficult to ensure.

of a tortuous kind which may happen to be in existence.

Fig. 92 shows diagrammatically the arrangements of the high-tension circuit. The low-tension circuit is of the customary form, except that the trembler shown in Fig. 93 is of an extra sensitive form. The distributor is placed in the high-tension circuit. The makers of this ignition system claim that owing to the adjustments made no possible error in the time of firing can arise which exceeds $\frac{1}{3000}$th part of a second. Also that in virtue of the nature of the spark the system is particularly suitable for use when the fuel used is one of the heavier brands of petrol, or paraffin or other heavy oil which may cause carbonaceous deposit on the ignition plugs.

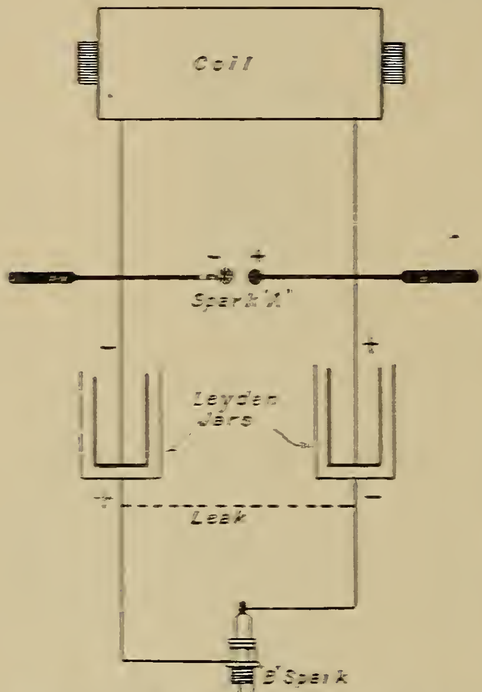


FIG. 92.—The Lodge Ignition System. Diagram of H.T. circuits.

163. Magneto ignition may be either *high tension* or *low tension*. No coil is used and no batteries or cells are wanted. The low-tension method proceeds on the principle that when a current is flowing it has energy of motion equal to $\frac{1}{2}LI^2$ (analogous to kinetic energy, $\frac{1}{2}mv^2$), and that if L, the self-induction, is made very great and I, the current, as great as convenient, the energy stored up is so considerable that a large or "fat" spark is caused to occur when the

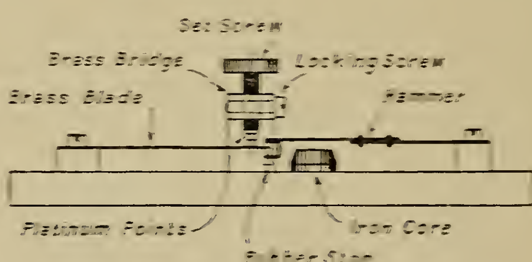


FIG. 93.—Lodge Sensitive Trembler.

circuit is suddenly broken. A low-tension magneto, or electric generator, is designed so as to cause such a current to be passing at the moment when ignition is desired to occur and, at the same instant, the circuit is mechanically broken in the cylinder and a spark passes. Fig. 94 shows diagrammatically the wiring for this system and the sparking plug used. A larger view of such a plug is seen in Fig. 95. A disadvantage of this system is the introduction of moving tappets into the cylinder, and the necessary provision of means for operating them from outside.

164. The High-Tension Magneto.—In this machine the current is generated by a shuttle armature which rotates between the poles of strong steel magnets. The rotation of this armature in the strong magnetic field results in the induction in its winding of an electrical current which is utilized for the purpose of ignition. The armature is wound in two parts, of which one is a primary winding, consisting of a few turns of heavy wire, and the other a secondary wind-

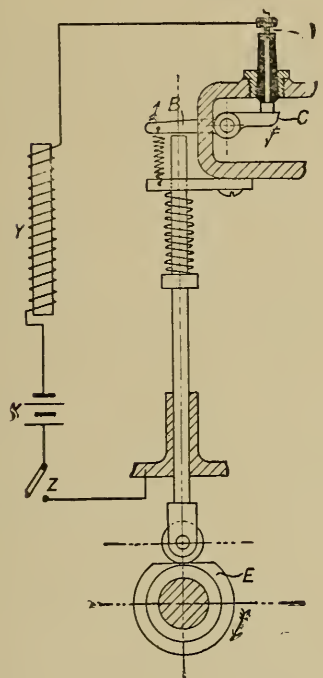


FIG. 94.—Low-Tension Magneto Ignition. *X* and *Y*, magneto machine shown diagrammatically. *A*, Spark plug. *C*, Contact point where circuit is closed and broken. *B*, Lever worked by rod running on cam *E*. *E*, Cam on half-time shaft. At the moment when the magneto is passing its maximum current around the circuit, the cam causes the circuit to be broken at *C*, so producing a spark at that point.

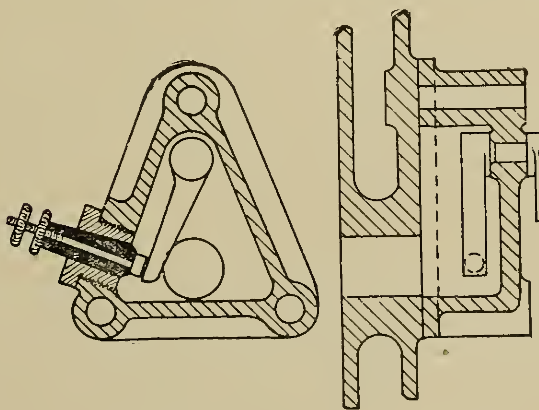


FIG. 95.—Typical Low-Tension Magneto Spark Plug. It will be noticed that this system of ignition requires moving contacts in the cylinder, which the high-tension system does not.

ing, consisting of many turns of fine wire. The effect is that a high-tension current is given off by the armature, as the design practically amounts to the inclusion in the armature of the windings of an induction coil. An outside view of this magneto is shown in Fig. 96, and its manner of working is shown in Fig. 97.

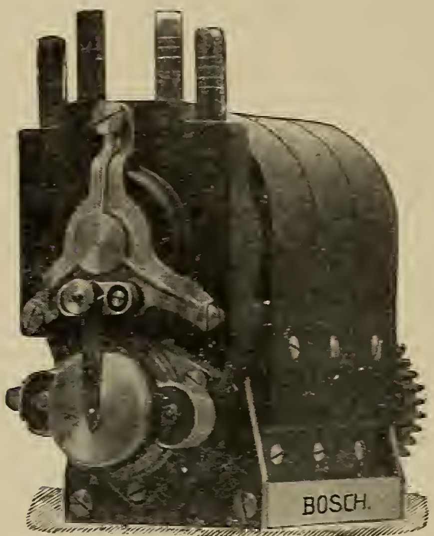


FIG. 96.—Outside view of Bosch H.T. Magneto. (A low-tension magneto is of generally similar shape.)

This system has the advantage that no moving parts need to be introduced into the cylinder in order to produce a spark. The voltage is so high that a "safety valve" spark-gap is usually fitted in parallel near the magneto in order to allow any unduly high voltage current which may be produced to pass across it. The spark in this gap is, of course, of no use except to act as a safety valve or bye-pass.

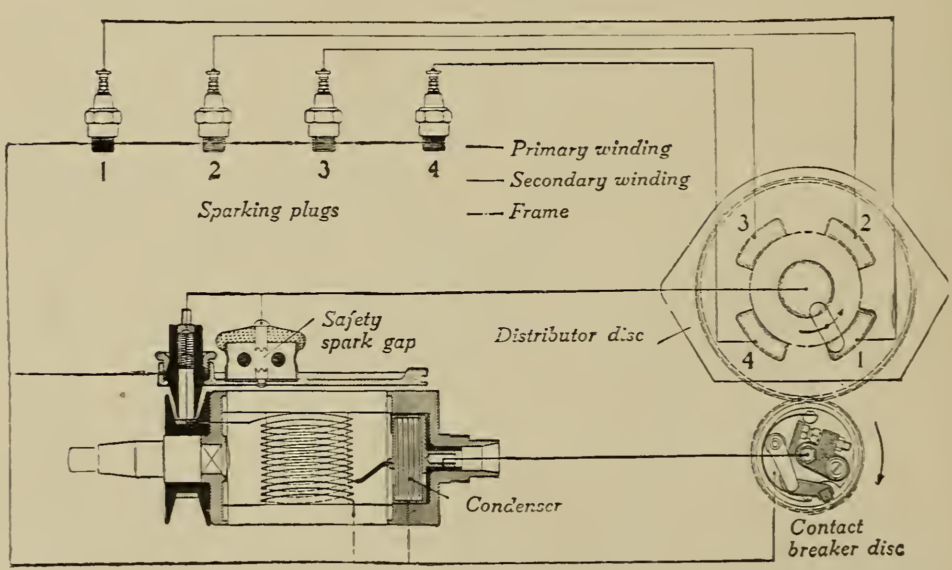


FIG. 97.—High-Tension Magneto Ignition, as arranged for 4-cylinder Engine.

165. Dual Ignition.—In a later form of high-tension magneto ignition, an accumulator and coil is added. The coil does its own making-and-breaking, but the distributor and the sparking plugs are common to both systems. In normal running the magneto is switched on; but when starting the coil is used. It is easy to tell which is in use, as the coil makes a buzzing noise. This system has become popular since the accuracy of manufacture of petrol engines has risen to such a high pitch. The fit of the piston, piston rings and valves is now so good that any compressed gas there may be in the engine on stopping will stay compressed for some hours and in most cases the engine will start from rest by merely switching on the coil ignition, so saving the labour of turning the engine round by the starting handle. This system, of course, is specially adapted for the petrol engines used on motor cars.

For gas engines, large or small, the ordinary high-tension or low-tension magneto is employed. Sometimes the ignition occurs at one fixed point in the stroke and sometimes it can be varied by hand or by the action of a governor. Some engines have two high-tension plugs in series so as to ignite the gas from more than one point and so produce a more rapid explosion. The ignition plug should never be put at the end of a recess or else a pressure wave may be produced which will cause *detonation* and possibly break the cylinder bolts and so lead to a bad accident.

166. Timing of Ignition.—One of the most careful adjustments of the ignition is its timing. That is to say, the regulation of the moment of sparking in the cylinder. If the spark is late the piston will have moved part of its outward journey, with the consequence that the effective working stroke is lessened and the mean pressure is lower than it need be. If the spark is too early, so that the gases are still being compressed when the spark comes, there is a knock in the cylinder when the explosion occurs. Normally the spark should occur just as the piston is at the top of its stroke, although since ignition takes a fraction of a second to spread throughout the mass of the gas it is necessary when the engine is running fast to time the spark to occur a little before the

dead centre so that maximum pressure is reached when the piston is just beginning its stroke. Engine speed is, however, not the only consideration affecting the timing; when running with weak mixtures the ignition takes longer than with rich mixtures so that to use a weak mixture it is necessary to "advance" the spark, i.e. make it occur earlier. It follows, therefore, that in the ordinary running of a car the ignition requires as much attention as the throttle, if the engine is to work at highest efficiency. An additional complication arises when coils having tremblers are used with batteries or cells, as the speed of "trembling" being naturally independent of the speed of the engine, it follows that at high engine speeds the sparking in the cylinder is apt to be somewhat erratic, sometimes coming early and sometimes late.

In a paper read by W. Watson before the Royal Automobile Club an interesting account was given of certain experiments undertaken to ascertain the character of the spark in relation to power. The engine used was a two-cylinder one, 3·5 in. × 4 in., with mechanically operated valves. The sparking plug was screwed into the cap used to close the hole over the inlet valve, the spark points being well inside a recess in this cap. The whole of the experiments were made on one cylinder only, the other being operated with a trembler coil and battery. The speed was 950/1,000 revolutions per minute. It has often been claimed that a "fat" spark improves the running, and that this was due either to quicker ignition of the charge or to more regular firing. Experiments with a trembler coil showed that although the weakening of the current was found to reduce the mean pressure, yet this *could be brought back to its original* value by advancing the spark. The result of this series of experiments was to lead Watson to the following conclusions—

1. As far as a petrol engine of the type used is concerned, the character of the spark which ignites the charge has no appreciable influence on the power developed.

2. With a trembler coil the time at which the spark occurs is liable to vary greatly, and on this account the power developed may be considerably reduced.

3. The variation in the time of firing obtained with trembler coils is different for different coils, and hence a multi-cylinder engine in which a separate coil is used for each cylinder is unlikely to develop its maximum power, particularly at high speeds; the reason being that although the tremblers of the coils may possibly be adjusted, for some particular voltage, so that each cylinder fires at the same point of the stroke, yet this adjustment will no longer be true if the voltage of the battery alters, particularly if it falls much below the value for which the tremblers were adjusted.

4. When a single coil is used in combination with a high-tension distributor, it is of very great importance that the current in the primary should never be allowed to fall to a value near the critical value for the particular coil. In this connexion it may be mentioned that, in Watson's experience, when the trembler is so adjusted for any given voltage of the battery, i.e. for a given current, that the note produced is very clear and "pure," then a very slight decrease in current, due to a small fall in the voltage of the battery, will cause the timing to be defective, owing to the region of the critical current being approached. Hence, with the normal current passing—i.e. with the battery fully charged—it is advisable to adjust the trembler so as to give a somewhat harsh and shrill sound, for then the current may be considerably reduced before the critical value is reached.

5. When selecting a coil, regularity in the working of the trembler for considerable variation in the current passing in the primary is of more importance than length or fatness of spark. Further, a coil taking a small current is to be preferred to one taking a large current, since trouble with the adjustment of the trembler blade will be decreased, owing to the reduced sparking at the platinum points with a small current.

6. Except for the fact that the engine cannot be started on the switch, the plain coil with a rapid break on the two-to-one shaft seems preferable to a trembler coil, since over a very large range of current—in fact, whenever the current is large enough to cause the passage of a spark in the cylinder—the timing is exactly the same. The advantage of the trembler

might be retained by using a switch, so that after the engine is started the trembler can be cut out, allowing the coil to act as a plain coil, a second condenser being provided.

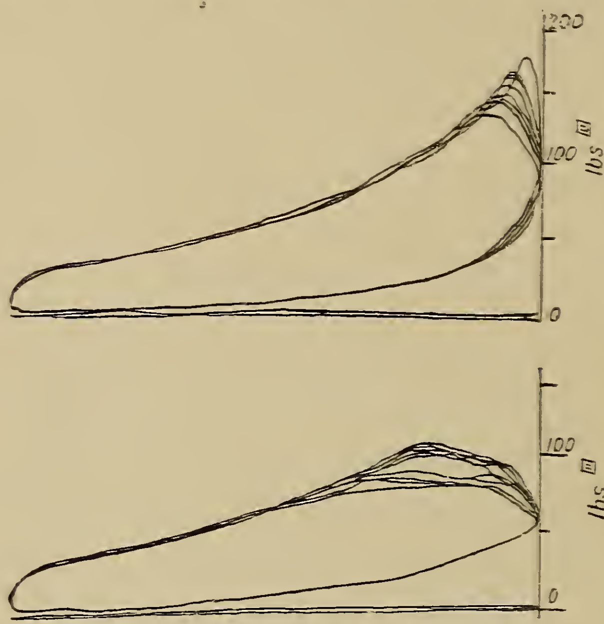


FIG. 98.—Indicator Cards obtained by Watson.

The two diagrams shown in Fig. 98, obtained by Watson, illustrate the advantage, so far as economy is concerned, of advancing the spark more than usual when employing a very weak mixture—that is, when driving with the extra air valve as far open as possible. The lower figure is that obtained when the spark is as much advanced as is advisable when using a full mixture, and the I.H.P. at 1,000 revolutions was 2·36. In the upper figure the spark has been considerably further advanced, so as to allow for the slow burning of a weak mixture, and as a result the I.H.P. at 1,000 is 2·76, an increase of nearly 17 per cent. in power, the consumption of petrol remaining the same.

EXAMPLES

1. An oil engine uses 0·8 lb. of kerosene per actual H.P.-hour; one pound of kerosene gives out 12,000 C.H.U. in combustion. What is the efficiency of the engine? (B. of E., 1911.)

2. In the test of an oil engine the analysis of the exhaust gases by volume gave: $CO_2 = 6{\cdot}8$ per cent.; $O = 11{\cdot}1$ per cent.; $N = 82{\cdot}1$ per cent. The oil analysis was $H = 15$ per cent.; $C = 85$ per cent. Find the excess air and the total mass of products per lb. of oil.

3. What weight of O is required for the complete combustion of 46 grams of alcohol (C_2H_6O) ? What weight of CO_2 will be formed, and what weight of water ?

4. The following data are taken from the test of an oil engine :—

I.H.P. = 65.
Oil used per hour = 45 lb.
Calorific value of oil = 19,500 B.Th.U.'s per lb.
lb. of air per lb. of oil = 70.
Jacket cooling water in lb. per min. = 68.
Temperature of cooling water, inlet = 62° F.
Temperature of cooling water, outlet = 138° F.
Temperature of exhaust gases = 430° F.
Temperature of engine room = 70° F.
Specific heat of exhaust gases = 0·24.

Draw up a table showing, as percentages, how the total heat of combustion is distributed.

5. In one pound of petrol there is 0·846 lb. of carbon and 0·154 lb. of hydrogen. What is the calorific value of petrol per lb. given that the calorific value of carbon is 8,130 C.H.U. per lb. and of H 29,100 C.H.U. per lb ? What weight of O is required for complete combustion of 1 lb. of petrol ? (B. of E., 1910.)

6. The area of a petrol engine diagram is (using the planimeter which subtracts and adds properly) 4·12 sq. inches, and its length (parallel to the atmospheric line) is 3·85 in. ; what is the average breadth of the figure ? If 1 in. pressure represents 70 lb. per sq. inch, what is the mean effective pressure ? The piston is 3·5 in. in diameter with a stroke of 4 in. What is the work done in one cycle ? If there are 800 cycles per minute, what is the horse-power ?

7. A petrol engine working on the Otto cycle has a cylinder 4 in. diameter and length of stroke is 4 in. The compression space is $\frac{1}{5}$ volume. Assuming the brake thermal efficiency to be 20 per cent., find the maximum power which the engine could give, running at 1,000 r.p.m. if at the end of each suction stroke the whole cylinder were filled with an explosive mixture of petrol vapour and air, having a mean calorific value of 57 C.H.U. per cu. ft.

8. A 4-cylinder petrol motor develops 60 B.H.P. at 1,500 r.p.m. What is the mean turning effect exerted on the crank shaft ? and what must be the ratio of the gearing between the engine and the driving axle, so that the car speed is 40 m.p.h. ? Assuming that the internal resistance of the car machinery is 20 per cent. of the power developed, what is the total external resistance against which the car is driven ? Diameter of the rear road wheels is 32 ins.

(B. of E., 1913.)

9. In a Diesel engine the compression ratio is 15·3 and the expansion ratio 7·5. The indicator cards give a nett I.H.P. of 201 and the

oil consumption was 67 lb. per hour, of calorific value 19,300 B.Th.U. per lb. Calculate the ratio of the actual thermal efficiency to the thermal efficiency of an ideal engine, receiving heat at constant pressure and rejecting it at constant volume, the compression and expansion being adiabatic and $\gamma = 1{\cdot}4$.*

10. In a motor cycle of 3 I.H.P. the mass of petrol used during a 4 hours' run at full speed is 8 lb. The highest temperature in the engine cylinder is 2,000° F. and air is drawn in at 60° F. Find the thermal efficiency and compare it with that theoretically possible. The calorific value of petrol is 18,600 B.Th.U. per lb.

11. The *Gnome* petrol engine develops 50 B.H.P. at 1,200 r.p.m. There are 7 cylinders each of bore 110 mm. The stroke is 120 mm. and each cylinder fires once in two revolutions. Find the average brake mean pressure (ηP).†

12. Air flows through an orifice from a reservoir in which the pressure is P lb. per sq. foot, and temperature T into a region of lower pressure, heat being neither received nor rejected during the operation. Obtain an expression for the maximum discharge in lb. per sec. in terms of P, T, the effective area of the orifice and the ratio of the specific heats. (Mech. Sc. Tripos, 1904.)

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* Cf. Ex. 32, on p. 43. † See par. 167.

CHAPTER IX

Petrol Engine Efficiency and Rating

Efficiency Tests under Various Conditions—Effect of Cylinder Dimensions on Power and Efficiency—Engine Rating—R.A.C. Rule—Callendar Rule—Operation of Two-Stroke Engine—Composition of Exhaust Gases as related to Efficiency—Motor Vehicle Tests.

167. Efficiency Tests on Petrol Motors.—Some of the most searching tests that have been carried out on petrol motors have been those undertaken in the Engineering Laboratory at Cambridge under Professor Hopkinson.

In one set of such tests * the engine used was a 16/20 H.P. Daimler four-cylinder engine capable of running at 250 to 1,400 revs. per min. Other particulars were—

Total volume of one cylinder with piston on out centre . .	0·04 cu. ft.
Volume of compression space .	0·0104 cu. ft.
Compression ratio . . .	3·85
Diameter of cylinder . .	3·56 inches = 90 mm.
Length of stroke . . .	5·11 inches = 130 mm.

The indicator used was a reflecting one of the piston type. The tests involved three sets of measurements—(1) engine losses, (2) B.H.P., and (3) fuel consumption. From (1) and (2) the I.H.P. could be obtained, and therefore the mechanical efficiency. The tests were run with the carburettor as fitted by the engine builders, and it must not therefore be taken that the engine was of necessity adjusted to give maximum power or efficiency.

* *Engineering*, February 8, 1907.

The results of the tests are shown in Fig. 99 in which curves are given for the I.H.P., B.H.P., the mean effective pressure and the torque on the crankshaft. The mechanical efficiency varied from 85 to 75 per cent.—falling slowly as the speed exceeded 600 revs. per min. The petrol used had a thermal efficiency on the lower scale of 17,000 B.Th.U. per lb., and on this basis the following table of thermal efficiencies was calculated—

Speed	Petrol Consumption (Pounds)			Thermal Efficiency	
Revs. per Minute	Per I.H.P. Hour	Per B.H.P. Hour	Per 1,000 Revs.	On I.H.P.	On B.H.P.
400	0·78	0·9	0·30	18·6	16·1
400	0·75	0·87	0·28	19·3	16·6
600	0·685	0·81	0·26	21	17·9
600	0·655	0·77	0·24	22	18·8
800	—	—	0·24	—	—
1,000	0·6	0·75	0·22	24·2	19·3
1,000	0·6	0·75	0·206	24·2	19·3
1,100	0·59	0·785	0·202	24·6	18·4
1,225	(0·65)	0·94	0·22	(22·3)	15·4

NOTE.—At speeds 400, 600, and 1,000, two tests are given to show the range of variation. At 1,225 the indicated horse-power is uncertain, as no direct measurement of loss was made at that speed.

The thermal efficiency rose considerably with increase of speed—due no doubt in part to there being less time for the explosive mixture to cool, but in view of the variability of the composition of mixture passed by the carburettor (of the usual jet type) it is not safe to build too much on these measurements. An interesting measurement in addition to the above was that of the pressure in the induction pipe. With the throttle wide open and a speed of 1,000 r.p.m. this pressure was about $1\frac{1}{2}$ lb./in.^{2} below atmospheric pressure. With the speed reduced to 400 r.p.m. this pressure was less than $\frac{1}{2}$ lb./in.^{2} below atmosphere. The mean effective pressure (P) in the cylinder was at its maximum value of 88 lb. per sq. inch when the speed was 600 r.p.m.; at this point the mechanical efficiency (η) was 85 per cent., so that the product ηP was 75. The expres-

sion ηP is in very common use in estimating the performance of petrol engines, and it is called the *brake-mean-pressure.* It is used in preference to P, the indicated mean effective pressure, because indicator diagrams of these engines are seldom taken (*see* Ch. V). With more modern engines than the Daimler engine on which the above tests were made the value of ηP would usually be about 90 lb. per sq. inch, and although in ordinary touring car engines this value would not be sustained beyond

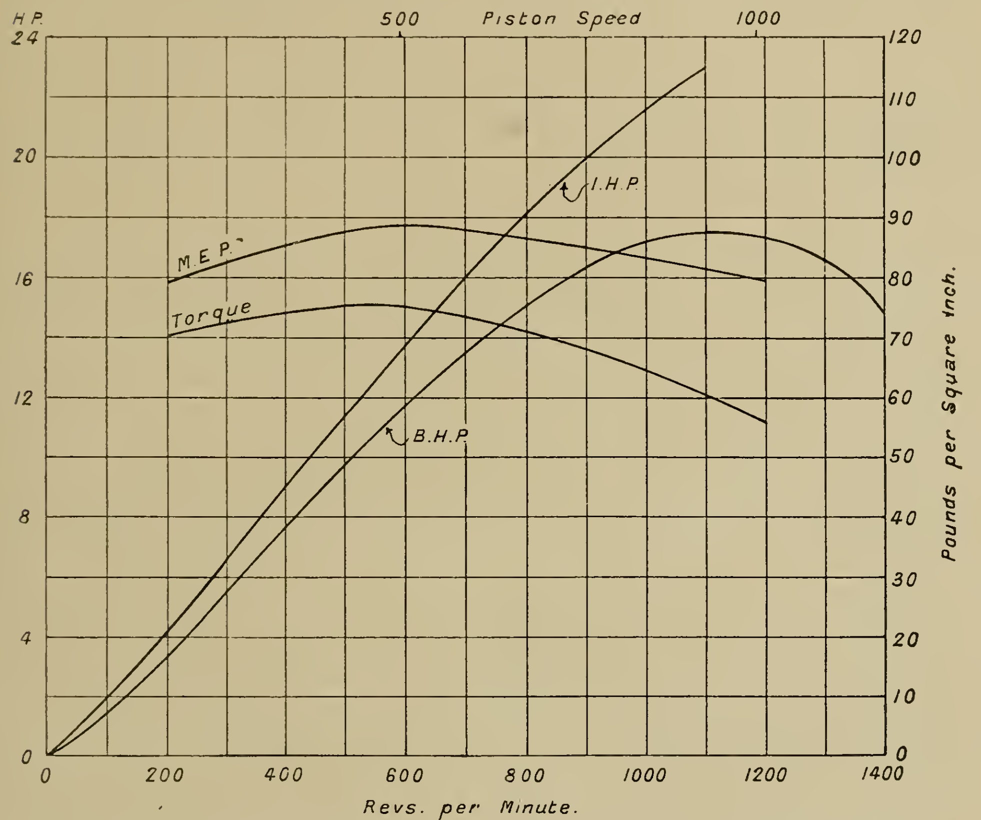

FIG. 99.—Professor Hopkinson's tests on 1/20 H.P. Daimler Engine.

about 1,200 r.p.m., in some special engines it is maintained till 3,000 r.p.m.

Tests * to discover the relation between the rate of consumption of petrol per B.H.P. and the character of the spark employed were undertaken at the City Guilds College by Messrs. Topham and Tisdall. A small single cylinder De Dion

* *Engineering,* December 28, 1906.

engine was used of 66 mm. bore, and 70 mm. stroke. The proportion of air to petrol varied with atmospheric conditions and was adjusted for each test so as to give a maximum power development, the throttle was kept fully open, and the air supply alone was varied. The engine was run at full normal speed. The chief results obtained were as follows—

Source of Spark (high tension)	Petrol Consumption in galls. per B.H.P. Hour
Accumulator with no external gap .	0·208 to 0·187
Magneto with external gap . . .	0·289 to 0·249
Accumulator with external gap . .	0·226 to 0·24

Not much can be gained from these measurements, though it would seem that different forms of high-tension ignition have a generally similar effect on the rate of fuel consumption, the accumulator having a slight advantage. The result of using an external spark gap in the high-tension circuit is worth recording. The plug insulation resistance was over 1,000 megohms cold, but after being used for running the engine for five minutes, it fell to about 6 megohms. At a dull red the resistance fell further to 2 megohms, and eventually sparking ceased entirely. On the introduction of an external air gap in the circuit, however, sparking was resumed, even when the plug was made bright red hot, and the resistance had sunk as low as 800,000 ohms. The experimenters considered that this showed that leakage through the porcelain of the plug might become quite large enough to stop sparking entirely, and that the effect of the introduction of the external air gap was to prevent the application of the voltage across the sparking points until the instant at which current began to flow in the high-tension circuit, the spark then being of the nature of a condenser discharge, and the leakage being considerably reduced in quantity.

168. The Effect of Cylinder Dimensions on Power and Efficiency.—Many attempts * have been made to produce a work-

* *Proc. I.A.E.*, 1907 and 1911.

ing theory of the effect of cylinder dimensions, particularly cylinder diameters, on the economy and power of internal combustion engines. Most of them have owed their origin to the competitive trials of motor cars in which the various vehicles are classed according to horse-power, and which therefore require that the figures given should be properly comparable. For large internal combustion engines the existence of such rules is not of vital importance, since the comparison of one engine and another depends upon so many other factors, and, moreover, organized competitions are unknown.

The first attempt to deal with this matter on a scientific basis is due to Professor Callendar, who read a paper before the Institution of Automobile Engineers on "The Effect of Size on the Thermal Efficiency of Motors." Cylinders as large as 14 inches were considered which, although common in gas engine practice, were well outside the range of motor car cylinders. This made the paper the more valuable in the general sense, although from the strictly motor car point of view advantage would have been gained had the theory been based entirely upon engine trials with cylinders nearer the customary motor car size. As it was, however, the paper presented a general theory not only applicable to motor cars but to larger engines also.

It is well known that the "air standard" of efficiency is higher than the efficiencies obtained in practice, and that the ratio of the latter to the former is commonly about 60 per cent. This means a deficit of 40 per cent. owing to some cause or other. What is this cause? The answer is, first, that the "air standard" of efficiency is a far higher one than any actual engine can ever achieve, owing to the fact that whereas the value of γ assumed in the "air standard" equation is 1·40, its average value for the actual cylinder gases at working temperatures, taking the increase of specific heat into account, would be more nearly 1·3.* This alone accounts for about 20 per cent. of the 40 per cent. apparently lost, and the remaining 20 per cent. is due to various heat losses such as jacket loss, radiation loss, etc. In general, therefore, it would appear

* See par. 66.

that the 40 per cent. loss is divided about equally between the two, though in Professor Callendar's view it would be more correct to put the unavoidable apparent loss due to the properties of the gases down as 25 per cent., and the remaining 15 per cent. to the loss of efficiency owing to heat losses during the operations of the cycle. It is clear that, as the volume of gas in a cylinder will be proportional to the cube of the dimensions, and the surface of the cooling walls proportional only to the square of the dimensions, doubling the size of an engine will halve the heat losses due to surface cooling. The fact that the larger engine will probably not run at so high a speed has little effect on this conclusion, as although the time the gases will have to cool will increase with diminishing speed, yet the diminished speed will lead to diminished scrubbing of the cylinder walls by the molecules of the gases, and so leave matters much where they were. It may therefore be estimated that the loss of efficiency due to surface cooling will be $\propto \frac{1}{D}$ where D is the cylinder diameter. Some losses, however, such as the radiation loss, do not follow this law of dimensions, though what law they do follow it is not yet known. Still, one of the most important losses has been shown to be proportional to $\frac{1}{D}$, and as an attempt at a working theory, there is no harm in grouping the losses together and putting them proportional to $\frac{1}{D}$. This is what Professor Callendar does with several sets of engine trials. One set is based on his own experiments on an engine with a cylinder diameter of 2·36 inches, and the others are drawn from the Report of a Committee of the Institution of Civil Engineers. From the combined results on all four engines as given below, he finds that the value of the constant a to be put in the expression—

$$\text{loss of efficiency} = \frac{a}{D}$$

actually comes out as 1·0 when D is measured in inches. So that loss of efficiency $= \frac{1}{D}$.

Designation of engine	C	L	R	X
Diameter of cylinder, inches	2·36	5·5	9·0	14·0
Loss of efficiency $\left(=\frac{1}{D}\right)$	0·42	0·18	0·11	0·07
Resulting efficiency figure $\left(1-\frac{1}{D}\right)$	0·58	0·82	0·89	0·93
Observed relative efficiency as compared with air standard	0·44	0·61	0·65	0·69
Ratio of last two lines	0·76	0·75	0·73	0·74

It appears that the value of the above constant a, viz. 1·0, was chosen so as to render consistent the figures in the last line of the above table.

In this way the relative efficiency of any engine is written down as $0{\cdot}75\left(1-\frac{1}{D}\right)$ and if the "air standard" efficiency for the degree of compression under consideration be called E, then the

$$\text{thermal efficiency} = 0{\cdot}75E\left(1-\frac{1}{D}\right)$$

Had the "air standard" been a standard really applicable directly to gas engines, the figure 0·75 would have been unity, so further simplifying the formula. As it is the above equation shows that even the largest engines cannot get nearer the "air standard" than 75 per cent. It is useful to compare this conclusion with the values found in par. 65. Callendar uses these results in obtaining his P.C. method of rating, to be described in par. 170. Before coming to that, however, it is necessary to consider the simpler R.A.C. rating.

169. R.A.C. Rating.—The rating recommended by the R.A.C., and adopted by the Treasury, is that originally suggested by Dugald Clerk, who proposed

$$\text{rated H.P.} = nd^2 \div 2{\cdot}5$$

where n is the number of cylinders and d the cylinder diameter in inches. Thus a four-cylinder engine of 4 inch bore would have a rated H.P. of 25·6.

This rating is tantamount to assuming that for a given mean pressure in the cylinder the piston speed in feet per minute

will be the same for all types and sizes of engine. Experiment has shown that the mean pressure is practically independent of bore and stroke, but there is uncertainty how far it is safe to assume that the piston speed is the same in all engines. This uncertainty is due to lack of clearness as to what the rated H.P. is supposed to represent. It might be any one of these—

(1) Maximum H.P. on the bench when running "all out,"
(2) Maximum H.P. on level road when running "all out,"
(3) Maximum H.P. when climbing the steepest hill climbable, or
(4) Average H.P. when running on roads in normal unimpeded service.

Now for (1) the R.A.C. rating method is probably the most correct, using, however, a lower constant than 2·5. For (2) the engine speed in r.p.m., and not the piston speed, is the more nearly constant factor, and H.P. is therefore roughly proportional to cylinder volume, and is approximately = volume in cu. cms. ÷ 100. For (3) the engine speed is commonly in the neighbourhood of 800 or 1,000 r.p.m., and the volumetric rating therefore applies here also, and we may roughly say H.P. = volume in cu. cms. ÷ 130.

For (4) there is little data available, but the R.A.C. formula probably fits it very nearly with the present constant. We may therefore make out the following table:—(C = total displacement volume in cu. cms.)

(1) Maximum bench H.P. $= knd^2$ where k is some constant
(2) Max. H.P. on road $= C \div 100$
(3) Max. H.P. on hill $= C \div 130$
(4) Average H.P. normal uninterrupted service on roads $= nd^2 \div 2{\cdot}5$.

170. The Callendar Formula.—The R.A.C. rating assumes that all sizes of engines are equally mechanically efficient when worked under the same conditions, which is not the case. If D^2 be multiplied by the expression $\left(1 - \frac{1}{D}\right)$ as proportional to the mechanical efficiency, the result is to obtain the expression $D(D-1)$ which should be used in place of D^2 in the

rating formula. In this way Professor Callendar suggests a "P.C. rating" (Petrol Consumption rating) of

$$\text{B.H.P.} = \frac{D}{2}(D-1)$$

the figure 2 being the suitable constant. The P.C. ratings and R.A.C. ratings for a number of cylinder diameters are given in the following table—

D	R.A.C. Rating (per cylinder).	P.C. Rating (per cylinder)
Inches	B.H.P.	B.H.P.
1	0·40	nil
2	1·6	1·0
3	3·6	3·0
4	6·4	6·0
5	10·0	10·0
6	14·4	15·0
8	25·6	28·0
10	40·0	45·0
20	160	190

These two formulæ are shown plotted in Fig. 100. The Callendar formula $\frac{D}{2}(D-1)$ gives the H.P. per cylinder.

The result of using the P.C. rating would be, to quote Professor Callendar :—" According to the R.A.C. formula, a four-cylinder engine with 2 in. bore and stroke (like the F.N. motor cycle engine) is rated at 6·4 H.P., and is equivalent to a single-cylinder of 4 in. bore. According to my experiments the four-cylinder of 2 in. bore could not develop much more than 4 H.P. under ordinary conditions, and would stand no chance against the single-cylinder of 4 in. bore. A four-cylinder of 3 in. bore is equivalent to a single-cylinder of 6 in. bore by the A.C. rating, but according to the P.C. rating, the single-cylinder would have an advantage in point of power of 25 per cent. A two-cylinder of equal power on the A.C. rating would have an advantage of about 12 per cent. over the four-cylinder, and a six-cylinder a disadvantage of about 10 per cent." Professor Callendar also remarks :—" An obvi-

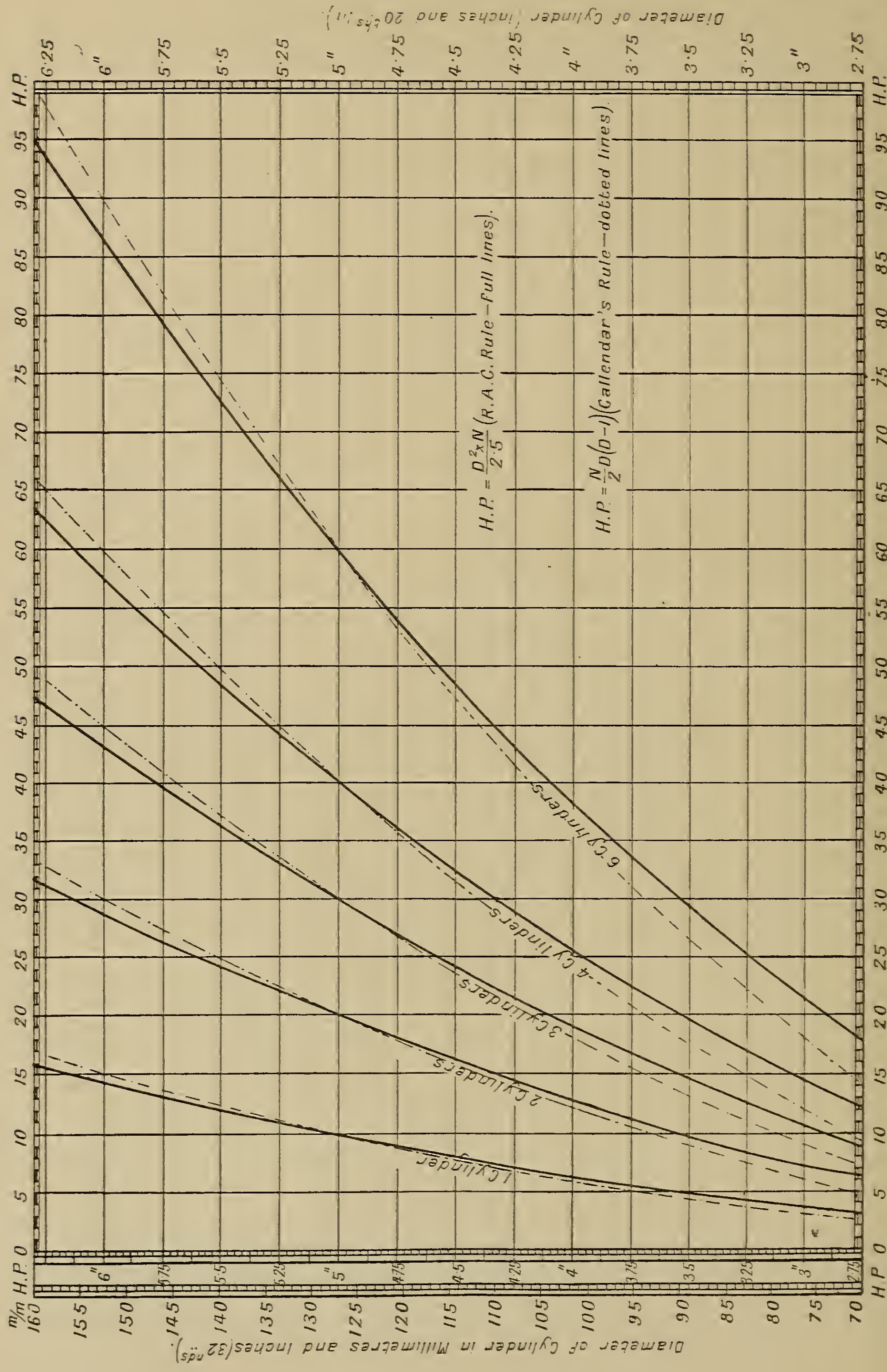

FIG. 100.—"R.A.C. Rating" for comparing the H.P. of Petrol Engines. Prof. Callendar's rule is also given.

ous objection to the P.C. type of formula is that the B.H.P. of an engine of 1 in. bore and stroke would be zero. According to the R.A.C. rating it should be $\frac{2}{5}$ H.P. It would no doubt be possible to get such an engine to run if very delicately made, but the effect of ignition lag would be serious at the normal speed of 6,000 revolutions per minute, and I doubt whether it could be made to give as much as $\frac{1}{10}$ H.P. on the brake."

At the present time there are few motor car engines that could not yield, without pressing, an amount of power given say by such a formula as :—

$$\text{H.P.} = \frac{\text{ND}(\text{D}+\text{S})}{5}$$

where N and D are as before and S is the length of the stroke in inches. If in a given engine the length of stroke be 10 per cent. greater than the diameter of the bore, this formula would attribute to the engine a H.P. greater by 5 per cent. than one in which stroke was equal to the bore. Whether the full engine H.P. can be utilized when the engine has been fitted to a motor car depends upon whether the gear-ratios have been suitably chosen.

None of these formulæ apply to anything but four-stroke single-acting engines. Some motor cars, however, have two-stroke single-acting engines, and for them the rating is low. A two-stroke engine can give, on the bench, about 60 per cent. more power than a four-stroke engine of the same bore, stroke and number of cylinders. The reason why two-stroke engines do not give twice as much as four-stroke engines is that the compression and explosion pressures are usually less, and there is often a considerable amount of fuel which escapes unburnt. When a two-stroke engine is fitted to a car a difficulty arises owing to its very powerful and noisy exhaust, the effective silencing of which increases the back pressure and largely neutralizes the gain of power there would otherwise be over the four-stroke engine.

171. Ratio of Power to Weight.—An interesting point is to find out what is the best cylinder diameter for minimum weight of engine per B.H.P. developed.

If the weight $\propto \text{D}^{2 \cdot 5}$ and the H.P. $8\,\text{D}(\text{D} - 1)$

Then $$\frac{\text{H.P.}}{\text{weight}}=\frac{D(D-1)}{D^{2\cdot5}}=\frac{D-1}{D^{1\cdot5}}$$ which gives the following table :—

D :—	1	2	3	4	5
$\frac{D-1}{D^{1\cdot5}}$:—	0	0·353	0·385	0·375	0·357

showing that the greatest H.P. per lb. weight of motor would be obtained when D = 3 inches, although D = 4 inches gives practically as good a result. The weight will however be affected also by change in *the compression ratio* since with increasing compression the engine parts must be made heavier.

172. Operation of Two-Stroke Engines.—Although the two-stroke petrol engine has twice the number of working strokes, per 1,000 r.p.m., that a four-stroke engine has, it suffers much from loss of power owing to some of the entering charge passing out of the exhaust before it has been burnt. Also the exhaust valve has to be opened before the end of the working stroke, and this diminishes the effectiveness of that stroke.

Careful experiments on two-stroke engines have been carried out by Watson and Fenning.* The engine tested was a single cylinder Day engine rated at $2\frac{1}{2}$ H.P. at 900 r.p.m., the cylinder bore and stroke being $3\frac{1}{4}$ in. When under test the power was absorbed electrically and the I.H.P. was measured by a reflecting indicator. Tests were made at 600, 900, 1,200 and 1,500 r.p.m.

The results are given in the following table (see next page).

The best values of P were obtained when the ratio of air to petrol by weight was 12 to 1, and the following were the figures :—

Speed r.p.m.	P lb. per sq. in.
600	63
900	58
1200	53
1500	48

These figures show how at high speeds there is not time for a full charge to enter the cylinder ; also that ηP would not be much more than half the value usual with four-stroke engines.

* *Proc. I.A.E.*, 1908 and later.

Test Number	Speed R.P.M.	I.H.P.	B.H.P.	Lb. of Petrol per 1,000 revs.	Lb. of Petrol per I.H.P. hour	Thermal Efficiency, Gross	Thermal Efficiency, Net.	M.E.P. lb. per sq. in.	Air / Petrol by Weight	Amount of Charge which escapes, per cent.
1	636	2·71	2·1	·0651	·917	·149	·230	62·6	11·24	35
2	596	2·51	1·9	·0660	·941	·145	·226	61·8	11·48	36
3	638	2·73	2·1	·0592	·830	·165	·252	62·9	12·45	34
4	609	2·48	1·9	·0531	·784	·174	·268	59·7	13·88	—
5	601	2·35	1·7	·0493	·756	·181	·279	57·5	15·12	—
6	604	2·36	1·7	·0493	·758	·180	·278	57·3	15·18	—
7	897	3·51	2·8	·0650	·997	·137	—	57·5	9·86	—
8	903	3·60	2·8	·0612	·923	·148	·212	58·5	10·35	30
9	902	3·59	2·8	·0579	·873	·157	·217	58·5	10·91	28
10	898	3·55	2·8	·0579	·881	·155	—	58·0	11·05	—
11	938	3·68	2·9	·0520	·795	·172	·247	57·7	11·76	30
12	905	3·54	2·8	·0541	·831	·165	—	57·4	11·83	—
13	900	3·57	2·8	·0493	·746	·183	—	58·2	13·02	—
14	937	3·55	2·8	·0452	·716	·191	·270	55·6	13·57	—
15	907	3·30	2·5	·0425	·702	·195	·276	53·4	14·86	—
16	897	3·26	2·5	·0426	·703	·194	—	53·4	15·31	—
17	1209	4·12	3·1	·0564	·992	·138	·169	50·1	9·75	18
18	1206	4·23	3·3	·0561	·960	·143	·177	51·6	9·91	19
19	1199	4·34	3·4	·0567	·945	·145	·175	52·8	9·92	17
20	1218	4·29	3·3	·0498	·847	·161	·201	51·8	10·96	20
21	1212	4·29	3·3	·0396	·671	·204	·250	52·0	13·87	—
22	1210	3·77	2·8	·0333	·641	·213	·262	45·8	16·74	—
23	1218	3·79	2·8	·0331	·637	·215	·264	45·7	16·89	—
24	1504	4·85	3·6	·0449	·838	·163	·179	47·2	10·06	9
25	1514	4·71	3·4	·0428	·825	·166	·177	45·7	10·43	7
26	1510	4·75	3·5	·0424	·808	·169	—	46·2	10·68	—
27	1509	4·80	3·5	·0398	·751	·182	—	46·7	11·29	—
28	1502	4·94	3·7	·0392	·716	·191	·206	48·3	11·61	7
29	1500	4·80	3·5	·0375	·703	·195	·208	47·0	12·12	6
30	1509	4·77	3·5	·0369	·700	·195	—	46·5	12·18	—
31	1510	4·82	3·5	·0342	·643	·213	—	46·9	13·10	—
32	1510	4·82	3·5	·0327	·614	·223	—	47·0	13·71	—
33	1508	4·91	3·6	·0325	·598	·229	·247	47·9	13·98	—
34	1508	4·51	3·2	·0304	·610	·224	—	43·9	14·75	—
35	1501	4·52	3·3	·0307	·610	·224	—	44·3	14·85	—
Throttle Valve partly closed										
36	639	2·25	—	·0486	·828	·165	·216	51·7	12·12	25
37	897	2·86	—	·0415	·781	·175	·187	46·9	11·27	6
38	896	2·89	—	·0416	·775	·177	·199	47·4	11·36	11
39	1208	3·55	—	·0339	·693	·197	·213	43·1	11·73	7
40	1510	3·65	—	·0269	·669	·204	·209	35·5	12·27	2

173. Composition of Exhaust Gases.—The efficiency of a petrol engine naturally depends on the degree to which combustion is complete. The exhaust gases should not, for good efficiency, contain any CO. All the carbon present should be burnt to CO_2. Nor, if the proportion of air is closely adjusted, will there be any oxygen in the exhaust.

It is difficult to write down the chemical formula in accordance with which the combustion of petrol takes place in an atmosphere of air, owing to the complex nature of the petrol molecule, but it is interesting to write down the combustion equation for C_8H_{18}, and to look upon it as representing, approximately, what occurs with petrol.

C_8H_{18} burns with O_2 as follows—

$$2C_8H_{18} + 25O_2 = 16CO_2 + 18H_2O$$

so that 27 volumes of mixture give 34 volumes of products, or, if the steam be condensed to water, 16 volumes.

In actual working, ordinary air and not pure oxygen is used, so that there is nitrogen also to be considered. With 25 volumes of oxygen, 94 volumes of nitrogen would be associated —making a total of 119 volumes of air. Each volume of this petrol therefore requires 60 volumes of air for complete combustion, and the equation can be therefore rewritten as—

$$2C_8H_{18} + 25O_2 + 94N_2 = 16CO_2 + 18H_2O + 94N_2.$$

The right-hand side of this equation is exhaust products, and the composition by volume will be—if the volume of the water be neglected—$\frac{16}{110}$ or 14·5 per cent. of CO_2 and 85·5 per cent. of N_2. If too little air were admitted some of the CO_2 would be reduced to CO, which being a poisonous gas is a very undesirable element in the exhaust; moreover, it would reduce the thermal value of the gas owing to a part of the carbon not being completely oxidized—a loss which has already been dealt with quantitatively in the chapter on suction producer gas. If too much air is admitted, free oxygen will appear in the exhaust. We have therefore the following three rules :—

(1) When oxygen occurs in the exhaust too much air has been admitted.

(2) Too little air leads to formation of CO.

(3) When neither CO nor O_2 appear in the exhaust the air is in the right proportion.

Some experiments on the composition of exhaust gases were made by Professor Hopkinson and L. G. Morse, in the engineering laboratories at Cambridge, on the Daimler engine already referred to.

The speed was kept at 700/750 r.p.m., and a jet carburettor of the usual sort was used. The throttle was kept open so that the suction never exceeded ½ lb. per sq. inch in the inlet pipe close to the inlet valves. Fuel used was Pratt's motor spirit; density 0·715 to 0·720; Calorific value 18,900 B.Th.U. (lower value). The exhaust gases were analysed by the ordinary volumetric methods, the CO_2 being absorbed by potash, the oxygen by pyrogallol, the CO by an acid solution of cuprous chloride, and the H_2 by palladianized asbestos.

The following table shows the results recorded—

EXPERIMENTS MADE BY PROFESSOR HOPKINSON AND L. G. MORSE.

Petrol consumption in lb. per 1,000 revs. . . .	0·181	0·191	0·197	0·217	0·250	0·293
Brake load at 43 in. radius lb.	25	27·5	29·3	29·4	29·3	27
Thermal efficiency . .	0·244	0·252	0·261	0·238	0·204	0·162
CO_2—measured . . .	10·9	12·8	13·5	10·6	9·6	6
O_2 ,, . . .	3·6	1·5	0·2	—	—	—
CO ,, . . .	—	—	0·7	5	6·25	11·6
H_2 ,, . . .	—	—	—	2·1	2·65	8·7
N_2 by difference . . .	84	84	84	81	80	73
Total O_2, calculated from N_2	22·4	22·4	22·4	21·5	21·3	19·4
H_2O calculated . . .	15·8	16·2	16·8	16·8	17·2	15·2

From this it will be seen that when the CO and O_2 are at a minimum, the CO_2 is 13·5 per cent. and the N_2 84 per cent., figures which are very close to those calculated above from the approximate chemical formula. Moreover, it will be seen that it is at this point that the highest thermal efficiency (0·261) was recorded. This is best brought out when the points are plotted in a curve, as in Fig. 101.

Curve B (thermal efficiency) shows how quickly the thermal efficiency declines when CO begins to be produced. Curve A

(corresponding to B.H.P.) is of a very different form, as although it is true that minimum production of CO corresponds to an output in H.P. very little less than the maximum, yet that maximum is found when the CO amounts to 0·7 per cent. and is very nearly maintained even when the proportion of CO rises to over 6 per cent. The ideal condition of working is

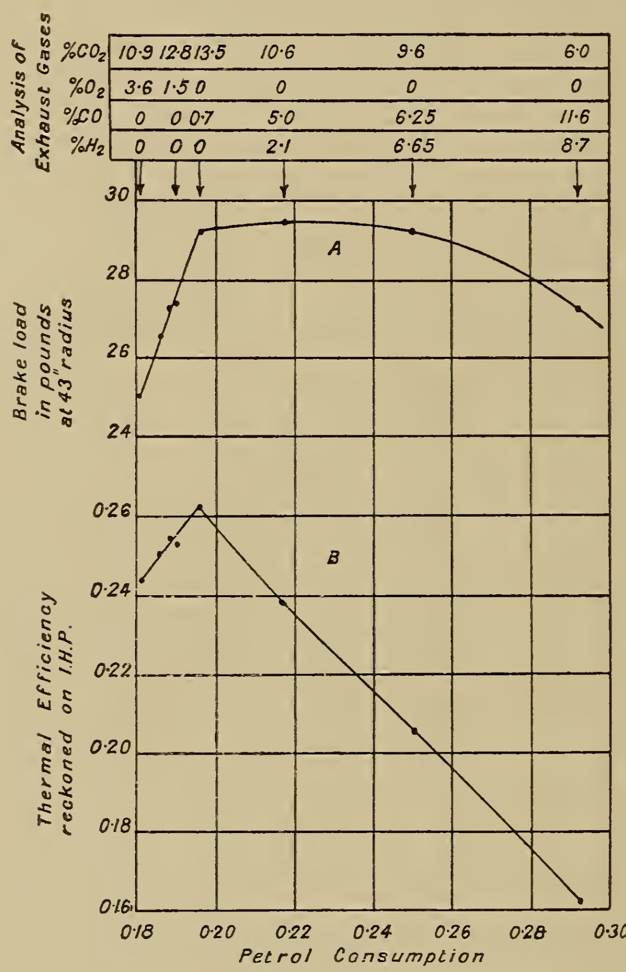

FIG. 101.

obviously the apex of Curve B, but the corresponding point on Curve A is not a convenient one to work at. In all engineering work it is customary to work, if possible, near to the middle of a curve which has a slow hump, such as Curve A, in order that small variations to right or left may not make much difference. The ideal working point on Curve A is

The following table shows the composition of the exhaust gases under these conditions on three different occasions—

EXPERIMENTS BY MR. DUGALD CLERK ON EXHAUST GASES FROM 18 H.P. SIDDELEY CAR. EXHAUST GAS ANALYSIS BY MR. BALLANTYNE.

	(a) Car standing; engine running as slowly as possible. No load.			(b) Car standing; engine running at about 600 revs. per min. No load.			(c) Car running on level about 18 miles per hour. Throttle less than half open.			(d) Car climbing a hill; engine running about 1,000 revs. per min. Throttle full open or over three-quarters open.		
	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3
	*	*		*								
CO	0·5	0·4	—	none	3·6	1·8	6·9	4·2	2·4	3·6	3·6	2·2
H	0·2	0·1	—	none	1·2	0·6	2·4	1·4	0·8	1·2	1·3	0·8
CH_4	none	none	—	none	0·3	0·2	0·9	0·5	0·3	0·4	0·3	0·3
Hydrocarbon vapours	trace	trace	—	trace	trace	trace	trace	trace	trace	trace	trace	trace
CO_2	5·4	6·0	—	6·3	10·8	5·6	9·9	11·5	11·0	11·7	12·2	11·8
O	11·8	10·2	—	11·2	2·2	10·6	0·3	none	2·2	0·1	none	0·6
N	82·1	83·3	—	82·5	81·9	81·2	79·6	82·4	83·3	83·0	82·6	84·3
Totals	100·0	100·0	—	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0

* Engine had been running for some hours. Engine in other cases had been just started from all cold.

therefore by no means the most convenient practical point, which in this case would correspond to about 5 per cent. of CO. This temptation has led engine builders to set carburettors so to give maximum power, instead of aiming at minimum production of CO and therefore maximum thermal efficiency.

Dugald Clerk has also carried out tests of this kind. He used for this purpose the engine on a 18 H.P. Siddeley car. The engine was a 4-cylinder one, bore 4 inches, stroke 4 inches. Samples of the exhaust were taken while—

(*a*) The car was standing on the level with the engine running as slowly as possible.
(*b*) The car still standing, but engine running at about 600 r.p.m.
(*c*) The car running on a level at about 18 m.p.h., the throttle less than half open.
(*d*) The car climbing a hill, engine running about 1,000 r.p.m., and throttle from three-quarters to full open.

It will be seen from the above table that under circumstances which might quite often occur in practice about 4 per cent. of CO is being produced. To meet the difficulties of designing a carburettor which should mix air and petrol in constant proportions under all conditions of load and speed is no easy thing, indeed most builders aim at quite different mixtures, viz., those that make for ease at starting, for rapidity of "picking up," and other features of car management that make for ease of manipulation.

Dugald Clerk concludes from the results of his experiments that the following conditions appear to produce imperfect combustion—

1. Too rich mixture with insufficiency of oxygen.
2. Too weak mixture with excess of oxygen, but too slow a rate of ignition and combustion.
3. Irregular mixture—mixture supplied too rich in composition at one part of the stroke, and too weak in another; that is, bad mixture.
4. Engine and carburettor cold. This tends to cause imperfect combustion, due partly to low temperature and partly to bad carburetting.

5. Improper timing of ignition, and missed ignitions.

6. Igniting in the body of the cylinder, instead of in a port. This is liable to produce imperfect combustion at light loads.

The sixth of the above conditions is an exceptionally interesting one. It seems incontestably to be the case that the presence of "pockets" in cylinders leads to loss of efficiency, "pockets" being the name given to any recess in the top of a cylinder which has the effect of increasing the clearance volume. But it is almost equally certain that the presence of "pockets" improves the running of the engine under working conditions: and renders it, as it is termed, "more flexible." Pockets lower the efficiency, as they increase the ratio of surface to volume, but they render ignition more certain, as, even at light loads when the exhaust products left in the clearance space dilute the incoming charge very considerably, there is the likelihood of the neighbourhood of the sparking plug (which is probably situated in or near one of these pockets) being rich locally in explosive mixture, so ensuring the proper starting and timing of the ignition. Too much "pocketing" on the other hand may lead to detonation of the charge.

174. Tests on the Road.—Not only are petrol engines for motor vehicles tested on the bench, but also when fitted into their chassis. Such tests are kept as closely as possible to normal working conditions. Measurements of speed and H.P. are needed, also wherever possible tests of acceleration and of hill-climbing ability. To measure the B.H.P. on the road it is necessary to know the speed and the resistance to motion (R) in pounds per ton.

Then

$$\text{B.H.P.} = \frac{\text{Velocity in m.p.h.} \times \text{R} \times \text{weight of car in tons}}{375}$$

The speed is usually read on a "speedometer" and the resistance by an "accelerometer." Such an accelerometer is shown in Fig. 102. Its action * is that of *weighing* the forces

* For a description of the mechanism, and of tests made with it, see *The Engineer*, September 16, 1910, and *Proc. I.C.E.*, Vol. 188; also *Application of Power to Road Transport* (Constable & Co.).

which oppose the motion of the car, and the needle points to the figure on the scale showing the number of "pounds per ton" of resistance at any moment. Such devices are called accelerometers because they were first introduced for the measurement of train acceleration. They can however be used for other purposes.

175. Fuel Consumption Tests.—It is also usual to measure the amount of fuel used on a road-test, and to put the result

FIG. 102.—Accelerometer.

into the form "gross-ton-miles-per-gallon" of fuel, i.e., the product of miles run per gallon by the total moving weight in tons. This figure affords a useful comparison of car with car provided that care is taken to measure during the run the average amount of the resistance overcome by the method described in the previous paragraph. This is necessary, as with heavy vehicles the resistance is sometimes twice as high on winter roads as on summer ones, and the "gross-ton-miles-per-gallon" figures will be changed in proportion. It is best to express the relationship in the form "gross-ton-miles-per-gallon on roads of normal road resistance," and to

take 70 lb. per ton as a common standard of resistance. This brings all such tests to the same basis of comparison. Not only does the state of the roads affect the resistance, but the speed at which the car runs during the trial also affects it considerably. At high speeds the component of the resisting force due to air resistance is very greatly increased. On good roads the total resistance with the average four-seated touring car * is usually about

$$R. \text{ in lb./ton} = 50 + 0{\cdot}06V^2 \text{ where V is in m.p.h.}$$

For a typical motor wagon under the same conditions

$$R = 50 + 0{\cdot}10V^2$$

but much depends in either case on the form of the body of the vehicle, and experiment is the only safe guide.

176. Windage Experiments.—S. F. Edge once carried out tests at Brooklands to see what effect the raising of a large wind screen would have on the speed of a large Napier car (38·4 H.P. R.A.C. rating). His results were—

Area of Wind Resistance Screen in Square Feet	Speed in M.P.H.
30	47·85
28	50·0
26	52·9
24	56·15
22	54·0
20	55·5
18	57·0
16	57·6
14	60·0
12	62·5
10	64·2
8	66·15
6	70·25
4	75:0
2	73·8
0	79·0

These figures are shown plotted in Fig. 103. Students will find it interesting to estimate what the resistance law must have been to give these results.

* Vide *Application of Power to Road Transport* (Constable & Co.).

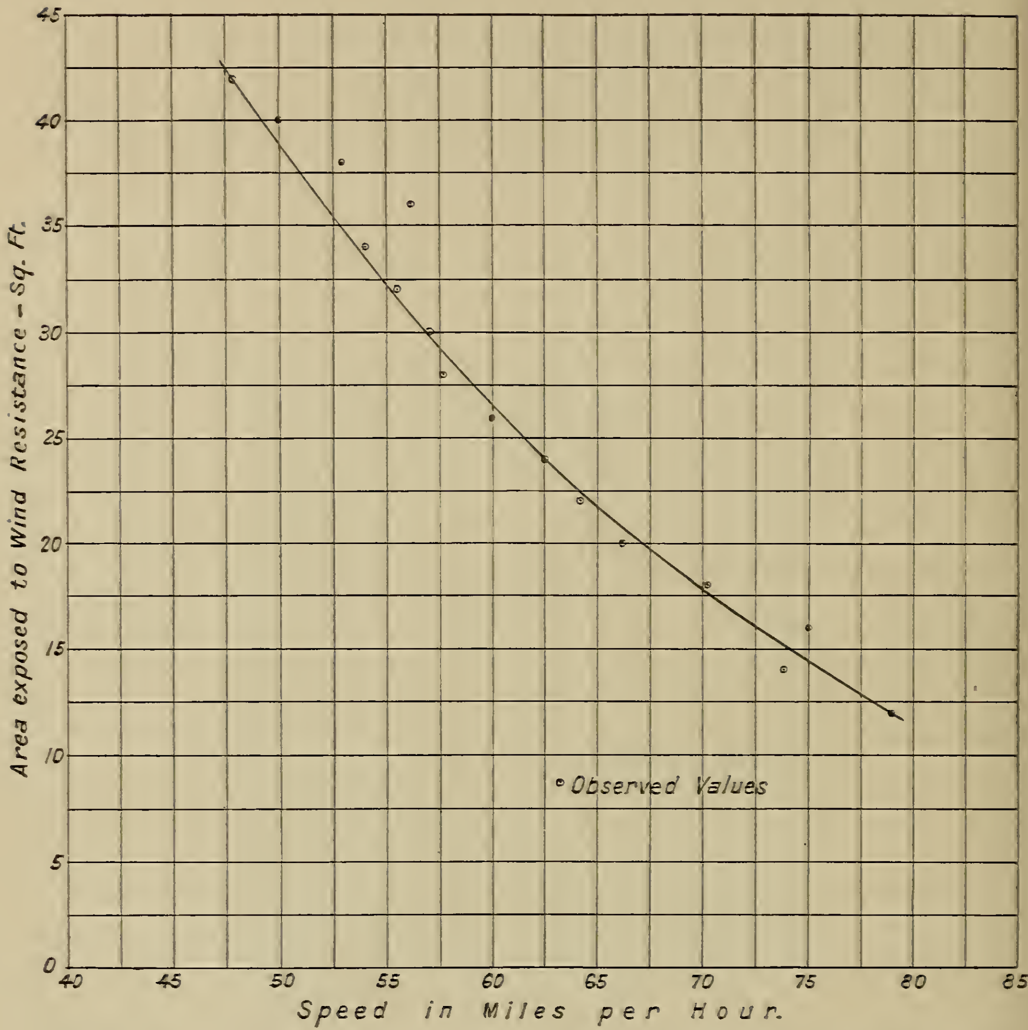

FIG. 103.—Diagram showing the effect of wind resistance on speed, as found by Mr. Edge on a Napier car carrying a special wind screen.

177. Hill Climbing Tests.—The initial acceleration with which a car can start to move on a level road is closely connected with the steepness of the hill it can climb. If α be the angle of the steepest hill climbable, then the equivalent acceleration is $g \sin \alpha$. This is obvious, as the same engine effort is needed to climb a hill of this slope as to give an acceleration equal to $g \sin \alpha$. Change of gear affects hill climbing ability in the manner illustrated in Fig. 104. The lower continuous curve is the resistance curve, whilst the continuous curve crossing it is the engine torque curve (on top gear) plotted to

the same scale, i.e., so that both show the equivalent tractive effort at the rear road wheels. The two dotted curves parallel to the resistance curve are the curves of resistance when ascending gradients of 1 in 20 or 1 in 12 respectively, since they represent the result of adding the extra effort necessary for hill climbing to that needed to overcome the resistance on the level. The upper dotted torque curve is that corresponding to the engine being on the next lower gear.

These curves show that on a level road the tractive effort

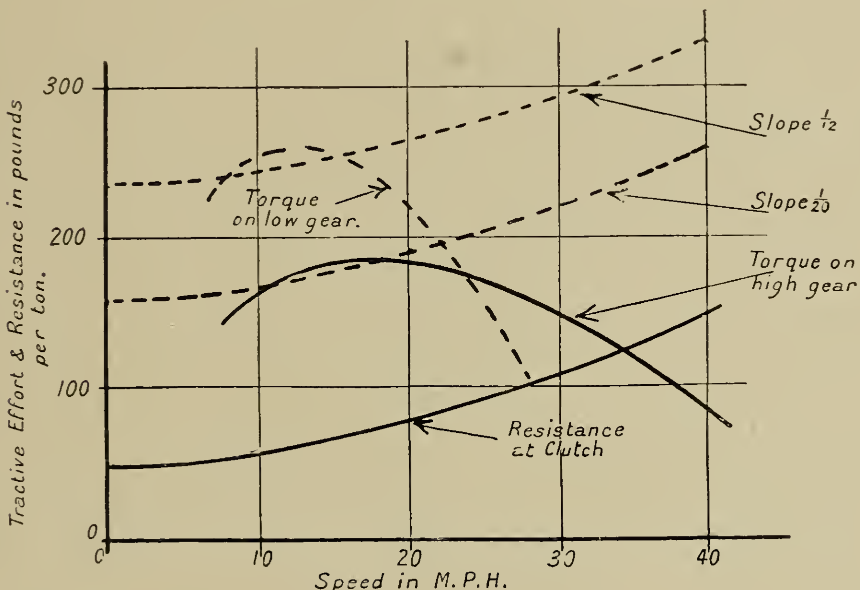

FIG. 104.—Tractive Effort and Resistance of a 15 H.P. motor car at various speeds.

and the resistance will be exactly balanced at 34 m.p.h. If the road began to ascend, the resistance curve would rise bodily upwards, keeping parallel to itself, and the crossing point of the two curves would move nearer in showing a lower speed of travel. At the crest of the torque curve the motion becomes unstable, and the engine will stop unless the "gear" be changed. This point evidently comes just after the slope rises to a steepness of 1 in 20. On the lower gear the point of instability comes soon after the slope exceeds a 1 in 12 gradient. These torque curves correspond to the throttle being wide open. If, however, the vehicle were travelling on

a level road on top gear at 20 m.p.h. and the throttle were suddenly opened wide, it will be seen that the engine would then be giving much more than twice the effort necessary to overcome the road resistance, and the balance would go to accelerate the motion of the car. In this particular case the excess effort is about 105 lb. per ton, and a force of 105 lb. acting on a mass weighing a ton would produce an acceleration of $\frac{105 \times 32}{2240} = 1{\cdot}5$ ft. per sec. per sec., which would, therefore, be the acceleration of the car at this point. It is by drawing curves of this kind that it is possible to design the best gear ratios of mechanically propelled vehicles.

178. R.A.C. Tests.—Interesting trials of touring vehicles and motor wagons have been organized by the Royal Automobile Club, and from some of their reports the following tables have been made out—

1907 TRIALS OF MOTOR WAGONS

Net Load carried	Average provided H.P. (R.A.C.) per ton of Gross Moving Load	Average G.T.M. per Gallon of Petrol
½ ton (5 cars) . . .	7·82	27·4
1 ton (4 cars) . . .	6·8	33·26
1½ tons (5 cars) . .	7·24	30·9
2 tons (7 cars). . .	4·96	34·93
3 tons (15 cars) . .	5·01	40·6
5 tons (1 car) . . .	4·87	33·9

R.A.C. INTERNATIONAL TOURING CAR TRIALS, 1908

H.P. (R.A.C. rating)	Miles run per gallon (average of best cars)	Average weight (loaded) of best cars. Tons	Gross-Ton-Miles per gallon of petrol. Average for best cars
Up to 20 H.P. .	24·2	1·30	30·7
20 to 40 H.P.. .	17·2	1·72	29·6
40 to 60 H.P.. .	15·9	2·14	33·9

Note.—Total distance run in the 1908 trials was 1,977 miles on the roads in Scotland and England. Average speed of all cars probably between 15 and 20 miles per hour.

These results, taken into consideration with many later ones the author has collected, suggest the figure of 50 gross ton miles per gallon as a convenient standard of performance when reduced to the common basis of a tractive resistance of 70 lb. per ton. This subject, however, needs a chapter to itself, and it is treated in fuller detail than is here possible in "The Application of Power to Road Transport."

EXAMPLES

1. Find estimates for the B.H.P. of the following cars, first on the R.A.C. rating of $\frac{D^2N}{2\cdot5}$, secondly using the formula $\frac{ND(D+S)}{5}$, D and S being the diameter of bore and stroke respectively, both in inches.

(i) 6-cylinder ; 5-inch bore by 7-inch stroke.
(ii) 4-cylinder ; 4-inch bore by 8-inch stroke.
(iii) 4-cylinder ; $3\frac{1}{2}$-inch bore by $4\frac{1}{2}$-inch stroke.
(iv) 2-cylinder ; 80 mm. bore by 280 mm. stroke.

2. If the B.H.P. of an internal combustion engine (four-stroke single-acting, one cylinder) be expressed by the formula $\frac{D(D+S)}{5}$, calculate the mean effective pressures which the formula assumes for the following mean piston speeds :—1,000, 1,250, and 1,500 ft. per min. of an engine having a stroke-bore ratio $\frac{S}{D}$ of 1·50, and a mechanical efficiency of 0·80.

ESSAY QUESTIONS

[The following questions are, for the most part, selected from examination papers.]

1. Explain what is meant by (i) absolute temperature, (ii) a perfect gas. State the two chief laws which perfect gases obey, and prove that for a perfect gas $\frac{PV}{T}$ is constant.

2. What is the law connecting the pressure (lb. per sq. ft.), volume (cu. ft.) and absolute temperature (centigrade scale) of 1 lb. of air ? [One cu. ft. of air at N.T.P. weighs 0·0807 lb.] Explain why the specific heat of a gas at constant pressure must be greater than the specific heat at constant volume.

3. A gas expands so that $PV^n =$ constant. Show that if n is the ratio of specific heat at constant pressure to specific heat at constant volume, the expansion is adiabatic. (Mech. Sc. Tripos, 1898.)

4. A gas engine works on an ideal cycle with adiabatic compression and expansion, receiving and rejecting heat only at constant volume. Obtain the expression of its efficiency. (Mech. Sc. Tripos, 1906.)

5. In what way does the PV diagram of the ideal cycle for a gas engine differ from reality? If it differs greatly, why are such calculations of any use? (B. of E., 1906.)

6. Describe with sketches the mode of operation of an Internal Combustion Engine. Explain why, in general, such an engine is more efficient as a heat engine than a steam engine of the same power. State where the various losses of energy occur. A gas engine of 10 B.H.P. consumes 180 cu. ft. of gas per hour, the calorific value of which is 690 B.Th.U.'s per cu. ft. Find the total efficiency, and give a rough estimate of the different proportions of energy lost due to the causes mentioned above. (Mech. Sc. Tripos, 1906.)

7. Describe a gas engine, and explain how it uses the Otto cycle of operations. Sketch the cylinder, showing piston, water-jacket, valves, shape of clearance space, and shape of exhaust outside the cylinder. (B. of E., 1899.)

8. What is meant by "scavenging" in relation to gas engines? How is it done, and how (or why) does it affect the efficiency? (Mech. Sc. Tripos, 1898.)

9. If the ideal P-V diagram of a gas engine consists of an area enclosed by two lines of constant volume intersected by two adiabatic lines, show that the efficiency of the cycle represented by the diagram depends on the compression ratio only, assuming that the specific heats of the working agent at constant pressure and also at constant volume, are constant. (B. of E., 1912.)

10. Explain why the efficiency of a gas engine falls short of the ideal value obtained by substituting $\gamma=1{\cdot}4$ in the formula $1-\left(\frac{1}{r}\right)^{\gamma-1}$

Indicate the relative importance of the different reasons. (Mech. Sc. Tripos, 1913.)

11. Criticise the Otto cycle from the point of view of (1) efficiency, (2) relation of power to weight on the part of the engine. In modern practice the tendency is to compress the mixture highly before ignition. How does this affect the points of your criticism?

12. How are Indicator-Diagrams taken from a petrol engine going at, say, 2,000 revs. per min? Describe the Indicator. (B. of E., 1911.)

13. Sketch the form of Indicator-Diagram you would expect to obtain from a petrol engine. Sketch diagrams showing:—

(*a*) The spark too much advanced.

(*b*) The spark insufficiently advanced. (B. of E., 1912.)

14. What ought to be the composition of the exhaust gases from a gas engine using good coal gas or from a petrol engine? Why does the actual composition differ from this? (B. of E., 1910.)

15. Make a careful sketch of a petrol motor. Show a carburettor to an enlarged scale and explain the principle of its action.
(B. of E., 1913.)

16. In a gas-engine diagram the expansion curve usually lies above the "adiabatic" expansion curve, showing that, if the working substance be a perfect gas, it must be receiving heat during the expansion; yet, in fact, much heat is withdrawn from the cylinder walls by the cooling water. What do you regard as the most probable explanation of this? Give some account of the arguments and experimental evidence which lead you to prefer your explanation to others that have been suggested. (Mech. Sc. Tripos, 1904.)

17. Discuss the reasons that have been given for the so-called "suppression of heat" in the working mixture of a gas engine, and give an account of recent investigations, conducted for this purpose, into the properties of the gas used and into the interchange of heat between the mixture and the iron surface into which it comes in contact during its working. (Mech. Sc. Tripos, 1911.)

18. Describe with sketches how lubrication of the various parts of an engine (not encased) is usually performed. (B. of E., 1902.)

19. Why do we regulate an engine with both a flywheel and a governor? Explain clearly how each affects the regulation.
(B. of E., 1900.)

20. Give an account of the different methods used for governing gas engines, stating the advantages and the disadvantages of each.
(Mech. Sc. Tripos, 1904.)

21. Sketch a section through the gas valve of a gas engine, showing the hit-and-miss mechanism operated by the governor.
(B. of E., 1907.)

22. Describe any non-luminous gas-making plant for use with a gas engine working to, say, 100 I.H.P. What chemical action takes place in the gas manufacture? What is the composition of the gas?
(B. of E., 1899.)

23. Describe with sketches the manufacture of any kind of producer gas. You must show that you have a knowledge of the chemical changes which occur. (B. of E., 1911.)

24. A petrol engine is run on the brake and the petrol supply is gradually increased by adjustment of the carburettor, the throttle being kept fully open and the brake adjusted so as to keep the speed constant. It is found that the brake load increases to a maximum and then keeps nearly constant, in spite of a considerable increase in the consumption of petrol. Also the maximum torque so determined diminishes as the speed is increased. Explain these observations. (Mech. Sc. Tripos, 1913.)

25. Describe some form of small 2-cycle petrol motor. Explain why the small 2-cycle engines do not as a rule give much more power than the 4-cycle engines of equal cylinder capacity and are considerably less economical of fuel. What countervailing advantages have

they when they are applied to the propulsion of vehicles and boats respectively? (Mech. Sc. Tripos, 1913.)

26. Why is the power of the engine of a motor car so great in comparison with the power applied to a horse-drawn vehicle? Give a rough estimate of how the power of a car rated at 38 H.P. is spent—

(1) When going at 30 m.p.h. on the level.

(2) When going at slow speed up a hill of 1 in 5.

What is the probable loss attributable to the crankshaft, the gear box and the back axle gear?

27. Show that the accelerations of the piston of an engine at the ends of the stroke are given by $\omega^2 r\left(1 \pm \frac{r}{l}\right)$ where ω is the angular velocity of the crank and r and l are the lengths of crank and connecting rod. (Mech. Sc. Tripos, 1912.)

28. The following are the results of two comparative tests of the same gas engine, the only difference between the two being that in test B the gas cock was opened wider than in test A, and the power was correspondingly greater:—

	A	B
Vol. of gas taken per stroke (cu. ft.)	0·10	0·13
Work done per stroke (per cent. of heat supply) .	30	27
Heat given to jacket water (per cent. of heat supply) .	29	34

The combustion of the gas was complete in both cases. What is the explanation of (1) the greater percentage heat loss in B, and (2) the lower thermal efficiency in B?

Is the greater heat loss adequate to account for the whole of the drop in efficiency, and, if not, how do you account for the balance? (Mech. Sc. Tripos, 1913.)

29. Show that the force required to accelerate the reciprocating masses of an engine is given approximately by

$$M\omega^2 r\left(\cos\theta + \frac{r}{l}\cos 2\theta\right),$$

where M = mass of reciprocating parts,

ω = angular velocity of crank,

r = crank radius,

l = length of connecting rod,

θ = crank angle, measured from line of stroke.

Give any graphical method by which the acceleration of the reciprocating masses may be obtained. (B. of E., 1912.)

30. Prove that there is no change of temperature when a perfect gas, after passing through a throttle valve, has been again brought to rest.

Experiments with hydrogen have shown a very slight rise of temperature. How would you account for this? (Mech. Sc. Tripos, 1912.)

31. In a single-cylinder engine, at any point of the stroke, show how to find the turning moment on the crankshaft, if we have an Indicator Diagram and the sizes and weights of the parts of the engine and its speed. Take the shortness of the connecting rod into account. (B. of E., 1910.)

32. Given the PV diagram of air altering in state, how do we find a diagram showing at every instant $\frac{dH}{dV}$, the rate of reception of heat ? If the expansion curve follows approximately PV^n = constant, find $\frac{dH}{dV}$. (B. of E., 1909.)

33. Explain how you would proceed to find the temperature of the charge in the cylinder of a gas engine at a point in the stroke just after the closing of the admission valve. Having determined this temperature, how would you use it to find the temperature at points during the expansion stroke ? State clearly the measurements you would make and the observations you would take to obtain the necessary data.

ANSWERS TO EXAMPLES

CHAPTER II

Pages 38–43

1. 1182° C. 2. 74·8 lb. per sq. inch by gauge.
3. 1870° C. 4. 1502° C.
5. 1598° C. ; 234 lb. per sq. inch.
6. 18·7 lb. ; 3,517 cu. ft.
7. (*a*) 58·8 lb. per sq. inch ; 70° C.
(*b*) 104 lb. per sq. inch ; 333° C.
8. 114 lb. per sq. inch. 9. 66·4 lb. per sq. inch ; 109·5° C
10. 330° F. ; − 63° F. 11. 500° C. 12. 30° F.
15. Work done in adiabatic compression = 65,400 ft. lb.
Work done in isothermal compression = 64,800 ft. lb
Work done by air in final process = 26,500 ft. lb.
Heat given to air = 92,000 ft. lb.
16. − 67·5° C. ; 37·5 cu. ft. 17. 406 C.H.U.
18. 13·6° F. ; 21·85 in. of mercury.
19. (1) 3,047 ft. lb. (2) 296 ft. lb. gained.
(3) 394° F. ; ·33 cu. ft.
20. 6·65 cu. ft. ; 440° C. ; 71·5 lb. per sq. inch.
21. 892° F. 22. 29 B.Th.U.
23. 0·566. 24. 75 lb. per sq. inch.
25. (i) ·0793 lb. (ii) 174·2 lb. per sq. inch ; 1185° F. abs.
(iii) 2523° F. (iv) 545 lb. per sq. inch.
(v) 46 lb. per sq. inch ; 1878° F. abs.
(vi) 49·4 per cent. (vii) 83·8 per cent.
26. (i) 1103° F. ; 1615° F.
(ii) On explosion 26,980 ft. lb. of Heat received.
On expansion 11,190 ft. lb. received.
On exhaust 25,690 ft. lb. rejected.
(iii) 32·5 per cent.
28. 1·79 : 1. 29. By compressor ; 20 H.P., 7·45*d.* per hour.
By direct heating, 1·8*d.* per hour.

CHAPTER III

Pages 64–65

1. 27 ft. lb.

CHAPTER IV

Pages 97–99

1. 107 C.H.U. 2. (i) $\gamma = 1{\cdot}37$. (ii) 7·55 cu. ft.
   ·00194 $(1{\cdot}37 - s)$ P C.H.U. per cu. ft. when the pressure is 5,000 lb. per square foot.
3. (i) 27° C. (ii) 177° C. (iii) 25·4 C.H.U. ; 62·4 C.H.U. received.
4. 102·5 lb. per sq. inch ; work done, 144,000 ft. lb. ; 103 C.H.U. carried away.

CHAPTER V

Pages 165–169

1. 6·7 H.P. 2. 20 H.P. 3. 94,700 ft. lb.
4. 81 H.P. ; 68·3 H.P. ; 84·3 per cent.
5. 666 H.P. ; 81·8 per cent. ; 91·9 cu. ft ; 112 cu. ft. ; 25·7 per cent. ; 21 per cent.
6. 56·6 H.P. ; 45·5 H.P. ; 80·4 per cent.
8. 45 H.P. ; 33·3 per cent. 9. 58·3 per cent.
11. 1·3 tons. 12. 155° C.
13. $2{\cdot}58 \times 10^5$ ft.-lb. ; $2{\cdot}65 \times 10^4$ ft.-lb.
14. 7·96 ft. ; 48 tons. 16. 39·2 ; 9·9 tons wt.
17. 466°C ; 1·57 ; 1·67 ; 1·81 ; 2·0 H.P.

CHAPTER VI

Pages 203–204

1. 126,000 ; 41,300 ; 42,000 ; 42,700 C.H.U.
2. 8,346 C.H.U. per lb. ; 11·56 lb. of air.
3. 25·3 per cent. ; ·46 per cent. lost in jacket-cooling water ; 79·2 per cent. lost in exhaust.
4. 1,002 C.H.U. per hour indirect heating ; 8,200 C.H.U. per hour direct heating ; 0·122.

CHAPTER VII

Page 216

1. 35·4 per cent. 2. 666 H.P. ; 81·8 per cent. ; 91·9 cu. ft. ; 112 cu. ft. ; 25·7 per cent. ; 21 per cent.

CHAPTER VIII

Pages 274–276

1. 14·75 per cent. 2. Excess air 106·6 per cent. ; exhaust products per lb. of oil, 32·1 lb.
3. O, 96 grams ; CO_2, 88 grams ; H_2O, 54 grams.
4. Indicated thermal efficiency, 18·9 per cent.
5. 11.360 C.H.U. per lb. ; 3·49 lb. of O.
6. 1·07 in. ; 75 lb. per sq. inch ; 240 lb.-ft. ; 5·8 H.P.
7. 8·8 H.P. 8. 210 lb.-ft. ; 3·57 ; 450 lbs.
9. ·655. 10. 20·5 per cent. ; 78·9 per cent.
11. 67·6 lb. per sq. inch.
12. $\dfrac{\cdot 819\, aP}{\sqrt{T}} \left(\dfrac{\gamma}{\gamma +}\right)^{\frac{1}{2}} \left(\dfrac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}}$
where a sq. feet is the contracted area of the issuing jet.

CHAPTER IX

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1. (i) 60 ; 72. (ii) 26 ; 38. (iii) 20 ; 22. (iv) 8 ; 18.
2. 105 ; 84 ; 70 lb. per sq. inch.

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